MODERN

MARINE ENGINEERING.

BY

N[icholas]. P[rocter]. BURGH, M.I.Mar.E., A.I.C.E.

With an Appendix, bringing the Information down to the Present Time.

LONDON:
WILLIAM CLOWES & SONS, Limited, 13, CHARING CROSS, S.W.
E. & F. N. SPON, 16, CHARING CROSS, S.W.
NEW YORK: 446, BROOME STREET.
1881.

MODERN MARINE ENGINEERING ILLUSTRATED

WITH

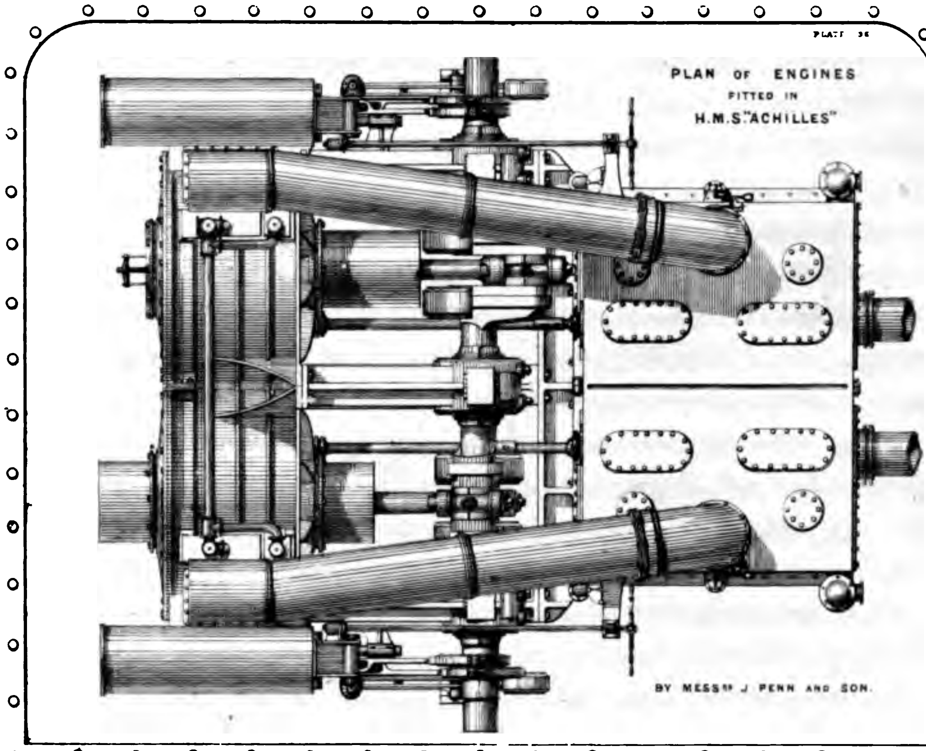
THIRTY SIX CORRECTLY DRAWN PLATES

AND TWO HUNDRED AND FIFTY NINE WOOD-CUTS.

BY

N. P. BURGH,

MEM. INST. MECH. ENG.



LONDON. E & F. N. SPON. 48, CHARING CROSS. NEW YORK. 446, BROOME STREET.

1872.

THE AUTHOR'S ADDRESS TO HIS READERS.

WHEN I commenced this work, I knew I had somewhat presumptively committed myself to a task worthy of the greatest authority on the subject. I was aware also that no *single* engineer could impart *all* the information. I therefore put myself in communication with the following firms:—Messrs. J. Penn and Son; Messrs. Maudslay, Sons, and Field; Messrs. James Watt and Co.; Messrs J. and G. Rennie; Messrs. R. Napier and Sons; Messrs. J. and W. Dudgeon; Messrs. Ravenhill and Hodgson; Messrs. Humphrys and Tennant; Mr. J. F. Spencer; and Messrs. Forrester and Co. The result is, that I have been enabled to give *bonâ fide* information, 259 woodcuts of details, and 36 coloured plates of examples in *actual* practice of the latest construction by the above firms.

For personal information I am indebted to J. Penn, Joshua Field, C. Barclay, T. B. Winter, and G. B. Rennie, Esqrs.; also my thanks are due to the leading gentlemen in their employ for their courtesy.

The "theory" and "practice," by all the leading firms, on *combustion*, *superheating*, *condensation*, *expansion*, and *propulsion*, have been put at my disposal, so that *all* that is stated on these subjects is the evidence of results.

The "engine-room fittings" have been fully explained and illustrated, also those relating to the boilers and funnels.

As the actual dimensions of the details of "engines, boilers, and propellers," are of the greatest importance to the young engineer, I have treated the subject in detail to the utmost extent.

On the *expansion* of steam and the utility of the *slide valve* for that purpose, I have endeavoured to throw some light.

The *strains* of the engine have also engaged my attention, so that the remarks and diagrams given are doubtless of relative value.

The *rules* I have given are deduced from scientific considerations, and I may add that a more complete analysis of this subject is in my "Pocket-Book of Rules," which fully agrees with the latest proportions.

The *weights* of the "engines," "boilers," "water," "coal," "propellers" "fittings," &c., in proportion to the "nominal horse power," are given in a tabular statement, which is taken from actual practice, and also the "duties of the engineers afloat" have been for the most part explained.

Before laying down my pen for the present, I may remark that throughout this work "facts" only have been introduced to demonstrate the *present* practice of the British marine engineers.

N. P. BURGH,

May 1st, 1867. 78, *Waterloo Bridge, London.*

MODERN

MARINE ENGINEERING.

CHAPTER I.

THE APPLICATION OF MOTIVE POWER FOR PADDLE AND SCREW PROPULSION.

PREFATORIAL REMARKS.

THE adoption of steam as a means for navigation was undoubtedly a great advance over that of manual and wind power, because the former is weak, while the latter is capricious.

Marine engineering of the present day may be said to embrace many branches. For the present purpose, that portion applying to paddle and screw propulsion will only be dealt with; for the fact that enough has been written on the entire subject, and volumes of historical facts have been published, causing the names of Farey, Tredgold, Rankine, and others to become generally known in allusion to the history of the steam engine, while the leading periodicals also, have not been free from the contagion of relating past events. In such works as those can be found every requisition to satisfy the most curious of historical science-searchers. This having been well digested by others, it will be sufficient for the present purpose to touch but lightly on the practice of even twenty years ago; for what seems to be required in the present day, is a work that shall carefully investigate the subject it treats of in modern detail; not alluding to each part singly, but to the many recognised actual improvements produced by the several makers. It may be said that previously this could not be attained, due to the fact that the manufacturing powers were slow in imparting or permitting information of practical utility. This want of confidence and idea of preserving much to gain *nothing*, is now happily becoming rare, and we find that the most eminent firms are the least suspicious of dishonour in those they trust. With that assurance of a certainty, this work will treat of facts; assumptions will only be introduced for the purpose of proving the poetry of natural laws, because for practical purposes a work should allude to practice in the main, for it is proved in every-day life, that every portion of our existence has a motive; and in like manner, each scheme or invention has its advantage.

The engineers and naval architects, that were once learned in stone and wood, bend with grace to the iron age of truth. Our forefathers sneered at the proposition of float-

ing iron steam-ships, and it is not long since that wise heads shook with condemnation at a vessel coated with nine inches of plating on her hull: therefore what was deemed a fallacy, has become a living illustration that *nature cannot lie.*

Of course doubts will always arise with novelties, therefore the positions of many of the most essential details of a marine engine are often unknown to the student; his knowledge extends perhaps to the cylinders and condensers, and he may doubtless know a little of the valve link motion, but there it ends. Little or nothing possibly he knows, too, of surface condensation and super-heating, while the many arrangements are entirely hidden from his conceptive powers; indeed, the commencement of a young engineer is generally an epoch in his life; filled with enthusiasm, he is ingenious in the extreme, which is worthy of a good cause. Doubtless the ideas of others being before him, prompts originality, and what is often the case, emanates discarded facts. These truths being in the possession of those above him, it is not too much to say that sundry cool receptions damp the ardour of the aspirant. Now those remarks are not out of place in a scientific work, because they literally pourtray the want of a compilation of what has been proved to be correct in marine engineering of the present day; for it is well known that in science, many productions can attain the same result.

Engineers in all cases have divided opinions as to design and proportion, and thus an honest rivalry is maintained. The object sought after in the present, as in times past, is mercantile, or rather commercial gain; let the trade or profession be what it may, opulence is the main cause. Certainly here and there are those who look on their studio, laboratory, or office, as the soul of their existence, and fame, the reward sought after. As a rule, however, commercial profitable employment is pursued with universal vigour by all engaged therein. This fact being natural, it is not surprising to find that engineers vie with each other in producing the most economical engine and boiler.

Influence in all transactions carries great success, so that it is not always the best maker or inventor who is meritoriously received by those in power. Knowing this, however, a statement of facts and representations of the latest improvements, cannot fail of appreciation from those who are interested in marine propulsion; and with this conviction, the present work will treat of all the latest improvements which are available, and direct attention to the fact, that to the present almost all the other marine books treat more of the past than requisite, with no attempt to give information that is of certain value. The intention, therefore, of this work now before the profession, in particular, is to impart that required information for practical utility. Each portion of the most important arrangements will be carefully explained and illustrated. The position of the cylinders for certain purposes will be liberally treated. The best arrangements will be fully described. Allusion in detail will be made to the many systems and arrangements for condensation. The various slide valves will be fully explained, and the value of expansion made

clear. The action of the slide valve and the link motion will be analyzed without exception. The remaining details will be descriptively illustrated, so that the student can value the cause and effect of the many productions. Paddle, single and twin screw propulsion, will be fully dwelt on. The correct dimensions in proportion to the speed required, and displacement, will all be investigated in order that the result shall be of utility. Boilers for high and low purposes will be described in detail. Super-heating and the causes of priming will in their turn be alluded to. Attention first will be given to the various types of engines and propellers.

PADDLE WHEEL ENGINES.

The application of natural laws gave rise to the steam engine. The anecdote of the little boy watching the rise and fall of the kettle lid, is perhaps the most truthful illustration of applied science that can be conceived. It doubtless occurred to the juvenile philosopher, that, if the bubbling of the heated liquid raised the lid of the culinary vessel,—by confining the agent,—an increase of power would be the result. Simple, and even absurd as this may seem, yet it was but a natural conclusion likely to emanate from a reflective mind. Step by step, although slow, has been the advance from the outset to the present; considering, however, the difficulties to be contended with, much has been done with credit to the pioneers engaged in the work of scientific development. At the commencement of the application of steam for marine purposes, the paddle wheel was deemed the only mechanical agency worthy of notice. Many other devices were promulgated to attain the most startling results. Paper and time were consumed in proving the gain to be ensured. Many offended originators withdrew from the scene of the scientific struggle, in high dudgeon at the want of respect, and belief, displayed by the Goths and Vandals of the age [*sic*].

Amongst other scientific eruptions, pumps were proposed, also windmill action applied to hydraulics was sworn to be the greatest attainment, if not the only correct mode. And in one instance a windmill action was proposed with the alteration in the action of the elements, viz., the vanes acted against the water, in the place of the air against the vanes; but at a considerable period after, the Sages of the age were stricken with the belief that the paddle wheel, or at least a similar principle, must be the correct, if not the most perfect mode of propulsion.

Now in order to test this assertion of well known facts, it will require a notice of a few of the many arrangements of the prime movers adopted, past and present. The position of the centre of propulsion of the paddle wheel is generally taken as the centre of the floats, and the centre of motion, of course, that of the crank shaft. Those positions were duly considered at the commencement, hence the arrangements of the engines were of a vertical adoption rather than horizontal. But the side lever engine is perhaps the father of all other devices, and indeed it may be truthfully said that in some instances it has outlived its progeny, and is doubtless an inverted beam engine, both in principle and practice; and for slow moving engines, per-

haps the best arrangement, always remembering, where expense of outlay and excess of space are not considered worthy of notice.

Parallel motion was resorted to for arrangements with the cylinders situated under the crank shaft. Engines of this class seem to have been favourably received; several makers vied with each other in attaining the best results. The air pumps in this case were worked by the outer end of the radius bar; and in some instances the bilge and injection pumps also received their motion from the same detail. With the vertical direct action it was found that short connecting rods were imperative. In some cases the parallel motion was dispensed with, and guides introduced. In another example, two piston rods were adopted, the crosshead having a projection inverted, thus a greater length of connecting rod could be attained; the cylinder cover, piston, and bottom end, were cast hollow to receive the consecutive projections.

The parallel motion direct acting engine, cannot ever claim much favour in the estimation of those who consider natural laws; for when the shortness of the connecting rod be taken into consideration, it is not surprising to relate that this type of engine was soon laid aside. There are, however, doubtless, a few persons that still advocate this arrangement, insisting that the parallel attainment is conducive of less friction than any other mode of guidance. To finally settle this argument will be to remember that the greater the length of the connecting rod, the lesser the angle of the same, and that all connected moving centres are alike affected; the strains imparted will be due to the angles assumed, and it is practically certain that a slight diversity in the length of the radius rods from the wearing of the bearings, will greatly affect the vertical action of the piston rod. A further cause for not advocating this class of engine is, that the radius rods are inclined to spring with high velocities, and thus shift the centre of rest, wear the centre of motion, and destroy the fixed points of length.

Direct acting engines, with guides on the top of the cylinder, have been, perhaps, more universal than those last alluded to; due to the fact that rigidity of truthful action can be better ensured. The double piston rod engine, with the inverted crosshead, was a step in the right direction, as far as the length of the connecting rod is concerned. The adoption, however, of two piston rods in the place of one, is an increase of packing, friction, and the extra requisition of attention and lubrication.

The next type of engine worthy of comment, is that which admitted of a greater length of connecting rod, while the depth of the vessel remained the same; this increase of the rod's length can only be understood by remembering that the height of the cylinder —when directly under the shaft—must be duly considered. In the place of guides over the cylinder, the sides of the same were deemed the better places. A crosshead connected to one or two piston rods, gives motion, by side rods, to two blocks working in suitable channels on each side of the cylinder; those blocks having pins projecting from the same, are attached to a forked connecting rod, and thus impart a revolving

motion to the crank shaft. This piston rod crosshead is, in most instances, guided by four guides, situated to clear the oscillation of the connecting rod; but the same length of rod can be attained with a more simple arrangement, by raising the cylinder from the base line, and adapting the standards as guides, a single piston rod, crosshead, and forked connecting rod, imparting the required action. Another arrangement also will produce the same result. The cylinder secured as before, but it and the piston, top and bottom cover, are annular, to allow sufficient space for the vibration of a single end connecting rod, but two piston rods in this case are imperative. In another example, to attain the required length of connection, the cylinder is secured at the base line, annular, as in the last exemplification; a T head crosshead from two piston rods imparts motion to a forked end connecting rod; the guides in this case are within the annular space in the cylinder. A similar movement has been effected by adopting two cylinders, and putting the connecting rod, end of the crosshead, guides, &c., between the former. Those two arrangements lastly noticed, are examples of the past; and the intention was undoubtedly to gain a greater length of connecting rod in proportion to those of other types, and such attainment was certainly the result. The first mentioned or side guide type, cannot be considered a judicious application of mechanical contrivances. Should the plurality of rods, guides, and crossheads, be properly considered, to say nothing of the weight of material, it will be conducive to condemning the arrangement entirely. Great preference cannot either be given to the raised cylinder type, although this may be said to be decidedly better than the first alluded to. Cylinders having annular spaces through them, are not the most economical kind, however the arrangement may deceive the imagination. Simplicity of connection is of course attained by this type, and a sufficient length of connecting rod ensured, but internal adjustment is imperative.

To design engines on land, and correctly manage, or rather, attend to them at sea, will puzzle many engineers, whose originations are nevertheless fair examples of their age of progression; because what may seem perfection of arrangement, even after construction, on land, will often betray want of foresight as to access for repair, or renewal at sea.

The double cylinder type cannot be termed economical, either for cost or space occupied; but the mode of imparting the desired action is not unworthy of comment.

Overhead motion for paddle engines is not much adopted in England in the present day, but the Americans still adhere to the arrangement of the beam being situated above the crank shaft. For smooth water and flat bottom vessels, there is not much objection to this arrangement, *i.e.*, as far as power is concerned, but for correct position of detail, doubtless, to be merciful is to be silent.

In the review now given, it must be strictly understood that strict chronology of diversity of arrangement is not adhered to, but merely a notice of those types which have been in actual practice, and thus proved their worthiness of notice.

Vessels of all classes are too often fitted

with masts towering like mighty representatives of the forest; it was no uncommon practice to provide,—when building a ship,—an addition for safety when on her beam ends; indeed it is not too much to say that the certainty of the casualty was more considered than the preventitude of the same. Now this may be said to have been rather a wise precaution, and to rail at it will be adverse to the time-honoured motto, that "Prevention is better than cure;" to reflect momentarily, however, will be to remember, that the absence of the cause for the *prevention* will ignore the requisition of the *cure*, and thus dispense with the two evils at the same time. The ballast for all vessels is best nearest the bottom of the hull, and indeed the individual who would propose otherwise, would be deemed "wanting" in the scale of science. This applies to examples of situation of machinery that have been sometimes seen, in practice, giving the lie in full to the correctness of natural laws. Happily, however, this want of respect to truth is now becoming lessened, and the marine engines of the present day are vastly superior to those of a quarter of a century ago.

The observations to the present have been strictly in reference to arrangements that are examples of the past, and allusion will next be made to the present and better improvements. The direct acting engine for the paddle wheel now deemed the best, is that of the oscillating kind. This class of engine at first met with many bitter enemies, and indeed it is not too much to state, that in more than one instance, the advocates themselves were in a state of doubtful oscillation of faith; the argument held forth in particular was, that the cylinder must become oval from the cause and effect; but engines of that class are now universal, and to the credit of the many makers it can be truthfully said, that perfection of construction is not far distant; the oscillating engine has nevertheless one great drawback, *i.e.*, friction of the trunnions; imagine a cylinder one hundred and twenty inches in diameter and ten feet stroke, and a fair idea can be formed: the friction is exaggerated, however, in some cases, and if simple calculation was only resorted to, the truth would be more apparent than it often is, in fact a doubt must always exist otherwise.

The general adoption of oscillating cylinders does not prove that perfection of arrangement is attained; for vertical action, the system is undoubtedly correct, and even then, care as to the balancing the parts in motion should be duly considered. Simultaneous almost with the appearance of the oscillating type, the single trunk engine presented itself. This class of engine, when first introduced, was fitted with oval trunks, sufficiently large only for the vibratory motion of the connecting rod; but there being a difficulty of shaping the external surface and fitting the same and gland respectively, the oval shape gave way to the circular kind. Trunks of a parallelogram in section have been proposed, and in some instances actually constructed, but with what result of course can be readily imagined; it will be better to add, that the correct packing of the stuffing boxes of engines of this class, is *the* great point to be considered, hence the universal

adoption of circular trunks by makers of those engines.

It has been argued here that position of detail in the hull should in all cases be duly considered; this can be readily understood to be correct, but the application of the law is the main point to be carried out. In all modern productions, the makers are careful to fix the bottom framing and cylinders as near the bottom of the hull as the floor will admit, which may be said to be a recognition of what every rivet boy knows. This, however, is not all that must be noticed: centre of motion, of the power produced, in relation to that expended, are matters which should in all cases be remembered, but as this chapter is merely a brief review of arrangements, with a silent opinion preserved, further remarks on those considerations will receive future attention.

PADDLE WHEELS.

The paddle wheel as a propelling agent, has met with a fair and protracted trial. Every scheme practicable might almost be said to have been tried. Floats have been made solid, others in two portions, while a third proposal has been to divide the floats into three separations. Fixed floats at the commencement were deemed imperative: after a while, however, it was conceived that a deal of power must be absorbed by the back water. But an ordinary land water wheel will convey perhaps a fair idea as an illustration; the action of the water turns the wheel from a point to a point, their positions to be considered as a regulator of power attained. Now on the water expending its power, it leaves the wheel due to its centre of gravity; reverse all these demonstrations, and a correct action of the wheel in the water can be known; in the one case the water glides from the wheel, while in the second instance the wheel leaves the element, but with a loss of power. The floats on leaving the water lift a certain portion of the same, but the greatest amount of friction is, directly after the floats pass the centre line, or when the same are deeply immersed. Many ideas have been exhausted to overcome these evils; feathering floats seem to be the most practical, although not giving out a correct action. The speed of the vessel, in all cases, determines the angles of the float on leaving the water; when it is remembered that the periphery of the wheel forms a series of curves intersecting below the surface of the water, or to the depth of the immersion, a just conclusion can be arrived at. In one instance loose floats have been proposed with the idea that the action of the water would regulate the required angle. This device has not met with much success, owing of course to the fact that the speed of the vessel in some instances negatived the action of the floats. Many arrangements have been proposed and carried into effect for the purpose of feathering the floats by mechanical action. To the present, the ordinary eccentric and radius rods remain paramount; in one case a crank has been proposed in the place of the eccentric action, but the principle remains unaffected.

What is actually required is, an unequal movement to be regulated by the speed of the vessel itself; the paddle wheel as a mode of propulsion, is undoubtedly correct for certain purposes, but for sea going vessels it certainly must give place to the later

production, commonly known as the screw propeller.

SCREW ENGINES.

The date of the invention of the common screw is doubtless unknown. Archimedes, according to history, knew its propelling properties, and the Chinese adopted it as a means of irrigation. As to the date of its adoption for the purpose of navigation, there are—as with most chronological events — divided opinions; suffice it to say, that in the present instance, practical evidences are deemed more worthy of notice than doubtful opinions past or present. It is not here hinted that a compilation is not of much or certain value, but rather that for the present purpose it is not requisite.

Arrangements of screw engines, like paddle engines, originated with the beam type. Next came the direct acting single-piston-rod class. Spur-gearing at the outset was deemed imperative. Many arrangements of tooth work soon became known. Divers schemes to preserve the point of contact, were published and experimented on. Position of the cylinder seems to have been the last matter considered; in some instances, what was carried out in principle by one maker, would be reversed in practice by another. Cylinders situated vertical, angular, horizontal, direct acting, oscillating, and rotary, have been proved, but as each arrangement is of course worthy of attention for the purpose of comparison, hence the description now following :—Vertical engines are of two or three kinds; the inverted cylinder arrangement can truthfully lay claim to the most universal adoption, due of course to the fact that direct action is attained without loss of space in the hull, therefore engines of this class have been variously situated. When spur-gearing has been introduced, the cylinders are secured on standards on either side of the centre line of propulsion; the diameter of the gearing of course in all cases determines the distance of the centres of the prime movers—it is almost needless to add that the space occupied is considerably more in proportion to that for direct acting engines.

The inverted engine rendered imperative that the power must be at the extremity of the arrangement. Now this was deemed worthy of improvement, which idea produced the annular vertical direct acting engine; engines thus arranged had the advantage of lowering the position of the cylinder in proportion to the stroke of the piston. This was of course a gain as far as detail was concerned, but with the entailment of the attendant evils of repair and internal friction.

We must here return to the position of the machinery in the hull of a vessel, in order to demonstrate practical exemplifications; because the correct method of arranging any class of marine engine is, first to observe the displacement of the hull. If it be required that a deep immersion be imperative, then arrange the engines accordingly; on the other hand, if an equal displacement must be retained, but shallow at the same time, an entirely different arrangement of detail is requisite. These simple but practical facts are not all, however, that need consideration from those who understand natural laws, because the class of engine to be adopted will be governed to a certain extent

by the transverse section of the hull: the correct form for which seems to be doubtful, inasmuch that the marine architects of the past and those of the present, disagree in relation to the lines of that portion. The primitive shape was that of a parabola, after which came the hyperbola, and we now find that the elliptical curve is being adopted in a compressed form. Flat bottom ships are also becoming much more universal than any other kind for large tonnages, so that more engine space transversely of the hull is available for better arrangements.

The type of engine known as that of the angular arrangement, was particularly adapted for hyperbolic shaped vessels. This class of engine was in most cases direct acting, while in some instances a return connecting rod arrangement has been preferred, or in principle as an ordinary table engine with side rods. The single and double trunk types have also been greatly used. The vertical oscillating type has had introduction, and in fact, both types have been well tested and applied.

Engines having their cylinders horizontally fixed in the hull, are now universal. This class, or rather arrangement, was not well received at the commencement; engineers were filled with the idea of putting the cylinder under or over the crank shaft; so that what had been a casual mode became a practice, and thus old fashioned ideas required eradication. The power of the engines up to the period now alluded to, was not large, and space was not available for alterations in most cases. Auxiliary steam power was in some instances only introduced, so repugnant was the new agent to seamen. Time and truth have, however, tended to dispel all doubts, and what was once an exception is now a rule; in passing it may be added that there are divided opinions even at present, as to what is the most correct arrangement of details.

Horizontal arrangements have had the advantage of having been well proven. Oscillating cylinders were first introduced, but the weight, increase of friction, unequal balancing, &c., soon proved defects which demanded attention. Single piston rod direct acting engines have also been well tested; but with engines of this last type, short connecting rods are essential, because with horizontal arrangements, space transversely of the hull must be noticed. Those facts doubtless gave rise to the introduction of the single and double trunk systems. With those the connecting rod can be increased to the centre of the length of the cylinder, or beyond that point if desirable; but when two trunks are adopted, space is required at the back end of the cylinder for the alternate projection of the trunk; and as this requisition is above the base of the cylinders, the *width* of the hull is sometimes unaffected. In order to lessen still further the space occupied, but retain the requisite length of the connecting rod, the double piston rods, return connecting rod engine was devised. This arrangement occupies less space than any other, longitudinally; but at the same time, has many disadvantages which the other types do not possess. The next class of arrangement now to be noticed, is that of the rotary kind. This type has met with many propagators, each of course being

[illegible]

[illegible]

[illegible]

stance, resigned in favour of the other in the next struggle.

Now to correctly define which propeller is the better, is to consider much of the cause and effect, and those who would decide hastily cannot be termed creditable authorities. The main facts to be noticed in either case are, the displacement of the vessel, tonnage per indicated horse power, area of immersion, consumption of the fuel, and lastly, speed attained in proportion to the total cost of maintenance, but those notations are not the whole of the many considerations requisite, but recognized in passing as the main. To describe *all* the attempts that have been made by the devotees to the screw system, would not impart much practical information; therefore, to give a brief account only of examples worthy of notice, will be of value for practical utility.

For the present purpose, three forms of propellers only will be alluded to, the Common, the Mangin, and the Griffiths. The first named was the result of a series of experiments, an accident determining the length on the line of keel. It was thought, at first, that the length of the screw should be extended to that of the pitch at least. Now on reflection, and at the best with facts for consideration, it is compatible with reason to imagine that, the friction and churning of the water will be more in proportion with a long, than with a short screw.

The shape of the blade, too, has met with much attention from different makers, cutting off the leading corner or edge, has been attempted as an efficient means to reduce the vibration at the stern.

With reference to the number of the blades, the shape and area will greatly decide that. We therefore now allude to a *common* double, treble, or quadruple threaded screw. Now taking the motion of a screw entering a piece of wood as the same as that of the propeller in the water, in principle a correct idea can be readily formed, with this difference. The screw entering the more solid matter has a less tendency for slip, or loss of advance, than when passing through fluid. To overcome that, elasticity of the blades has been tried, and screws thus made are universally termed the "boomerang propellers." It is held by the advocates of this type of screw, that by the blades springing at one part of the revolution, and recoiling at another stage, an additional power is attained, while at the same time a less vibration is the result; but the great evil to be overcome is the churning of the water, or in plainer terms, the separation of the same, for there is not the least doubt that the extremities of the blades are surrounded by a ring of fluid which extends towards the centre in the shape of a cone between the blades. And this theory is based on the fact that the displacement of the screw propeller is less at the boss than at any other part; while the progress of the screw is of course due to its pitch and diameter in proportion to the speed of the vessel, minus slip. As for immersion, many authorities advocate a deep draught for the propeller; others argue against that while specifying the evils, and urge a reduction of the same to gain power. Both partisans, however, are correct in their theories, for where a smooth surface of the element can be ensured, the depth for

the screw can be much less than for sea-going steamers. This is obvious when it is remembered that the motion of the surface of the water greatly lessens the effect of the screw as a propelling agent. As with sea vessels, the undulations of the waves greatly affect the immersion of the blades of the screw, hence the diversity of displacement in vessels for river and sea navigation. Much more could be said on the present subject in relation to the ordinary screw, but further observations will be better understood when assisted by illustrations shown hereafter.

The Mangin screw derives its name from its inventor. The principle of this propeller is a proposed attainment of less vibration at the stern post, and less churning of the water than with the ordinary screw, which has a uniform pitch. Now the Mangin form is, that the leading portion of the blade and the trailing part, are of uneven pitches, so that a section at the extremity of the blade would be angular rather than helispherical in plan, with four blades, or two two-bladed screws. The gain by this peculiarity of section is, that the water is supposed to be thrown off from the blade after leaving the centre line. This can be better understood by remembering that with the common helix section, the action of the blade is nearly equal throughout its revolution.

The Griffiths propeller, alike with the Mangin, owes its name to the inventor. The object attained in this example is much in effect as the former, with the exception, that the shape of the boss or centre part produces different effects. The boss end of the Mangin screw is usually abreast at the connection with the blade, while that of the Griffiths is globular. Now it is obvious that there is less resistance with a curved form passing through the water than that of "butt" shape. To better define this will be to assume the resistance between a wedge and a flat end shaped substance entering any body or fluid. This then is the variation in the bosses of the productions of the rival inventors. Both Mangin and Griffiths seem to agree on the main point worthy of consideration—that the effect of the blade in the water should obviate the churning of the water, although with different methods of attainment.

The Griffiths form of blade inclines forward in the direction of its action, or rather propulsion. The Mangin blade—alike with that of the ordinary screw—is of more width at the extremity than at any other part, and four blades used. The Griffiths blade is extended midway from the centre to the extremity, the lesser width being at the latter part. Next as to the correct speed of the vessel and that of the screw, much valuable time and money have been expended in testing the true proportion. Many critics speak against negative slip, and the effect of the same. Doubtless time might be better occupied in investigating the result of positive slip. Most engineers are aware that when the vessel moves faster than the theoretical speed of the screw, momentum must have been in force with a loss of power. On the other hand, when the screw gives out a speed of a given number of knots per hour, and the speed of the vessel falls short of the same excessively, *then* there is cause, and greatly too, for investigation to the point as to the

best proportion. Now the cause for these diversions of opinion seems to be questionable; some authorities lay the blame to the section of the hull immersed; others again reverse matters by coinciding with the neutrality of the displacement, but only investigate the pitch of the screw in proportion to the speed attained. Displacement of the hull certainly regulates the diameter of the propeller, but the pitch is regulated by the section of the hull immersed. This can be readily understood by remembering that the deeper the immersion, the greater the friction. To further define this, presume a ship, whose side immersion shall equal twelve feet on the curved line; then assume the depth perpendicularly to be two feet, and the length one hundred feet, then $100 \times 12 = 1,200$, at two feet draught. Now to exemplify further, presume the hull to be immersed four feet, and the area of immersion the same, it will be understood that the beam of the vessel must be contracted to retain that area, while at the same time a deeper draught is imperative.

From long and varied experiments it has been proved that the friction of bodies passing through fluids increases in due proportion to the depth of immersion; therefore the first advocates of ordinary propellers adhered to the fact of a fixed pitch irrespective of the displacement or power exerted. On reflection, however, it soon became apparent that a means of proportioning the pitch to the requisition would be a boon, and from this may be said to have originated the many schemes for altering the pitch of the blades; to produce which practically, two great matters required consideration—means of adjustment, and the correctness of the same. Bevil gearing, levers, wedges, and divers other acquisitions were soon introduced, each device of course claiming some merit. Amidst the promulgations at the outset, one great fact doubtless was forgotten, what would act on land was adverse at sea. Some inventors seemed to have discarded the knowledge that salt water causes an accumulation on all materials exposed to the same. Practice, however, with its always unerring effect, soon dispelled all doubts, and thus the ideas of the many gave place to the practical results of the few, which we will now explain as the best mechanical appliances.

The modes for regulating the angle of the blades to the required pitch, are now condensed into three kinds generally. The makers of the Mangin propeller, as a rule, regulate the pitch by slotted holes in the flange of the blade. The securing bolts and nuts cause a fixed connection, and the unscrewing of the latter allows the loosening of the blade.

The Griffiths screw, alike with the Mangin production, relies on the alteration of the angle, or rather the pitch of the blade. Rigidity being imperative, much depends on the means adopted to attain the same. Studs and nuts, or wedges and keys, have been much used, while in some instances the latter are dispensed with, and when those are not adopted, the locking of the stud heads or nuts is imperative.

The preventitude of looseness of the wedges and keys is attained by separate plates being secured by double nuts at the ends of the keys.

To further explain in detail the modes herein alluded to will be out of place, as for the present a brief allusion to past and present modifications is only intended.

TWIN SCREW PROPULSION.

The double mode of attaining the result of a single screw, as far as power is concerned, is now becoming less a novelty, but much has to be attained to produce similar economy with large steamers. As with all untried schemes, this mode was received by a double faction—the one side advocating and the other condemning. The first party held the system as a great boon. Amongst the many advantages set forth, a certainty of steering power is produced, and in the event of a break down of one portion at sea, the progress of the vessel can be continued with the other uninjured engines and screw. As for the gain in power for a given displacement, no definite conclusion has yet been made to warrant the assertion of economy of the twin over the single type for large sizes. For steering purposes and reservation of power, when required, the twin system has undoubtedly proved superior to any other, and the advocates of the scheme bring this forward as a crushing evidence of effect. With this mode of propulsion, however, as with most tangible inventions, doubtless much can be said in its favour; but it must not be supposed for a moment that *all* is gained by the adoption of two necessaries in the place of one. To argue fairly, will be to give each portion its just reward of attainment, without prejudice on either side. Now presume a vessel to be fitted with a single screw; again assume a similar hull having twin screws for propulsion, the displacement to be in proportion to the diameters of the screws; granting the speed attained to be alike, which must be the case if the same proportions are adhered to, what is the gain ? simply, a more certain effect for steering only. Doubtless, for war purposes, this attainment is invaluable; but to argue entirely against the single system, will certainly not be correct. For light draughts and river propulsion, the twin system is perhaps the better, but for sea vessels immersion must not be forgotten; the great gain in either case consists in the object to be attained. As before stated, steering power is invaluable for war purposes; but if the same cause be applied to mercantile vessels, the effect is greatly reduced. What is required in the one case as almost imperative, is scarcely worthy of attention in the other example. This argument will apply in all modes of scientific productions, each portion has its advantages for a given purpose. To eulogize and condemn, without justice, both at the same time on the same subject, can truthfully be termed prejudicial evidence; hence to correctly understand the question at issue is, to weigh the same in the scale of justice, and thus arrive at a correct conclusion.

The arrangement of the engines in the hull, for twin screws, forms no small feature to produce the best result; ready and certain manipulation being imperative as one of the advantages of the system. Many types of engines have been introduced for twin propulsion. The horizontal arrangement seems to be the most universal, while in some cases the overhead cylinder class is preferred. The

beam of the vessel will in every instance greatly affect the arrangement of the machinery; with contracted widths the arrangement is greatly disturbed, and one pair of engines situated in advance of the other, imperative, for horizontal arrangements.

With overhead cylinders a perfect twin distribution of details is attainable, but at the same time entails the evils heretofore alluded to. To decide on the best kind of engine will be dogmatical without alluding to the beam of the hull, and displacement of the same. Some authorities have proposed angularly inverted engines, arranged thus—the cylinders are secured on a frame centrally secured between the crank shafts, the condensers and pump being horizontal in action. Other makers prefer duplicate, return connecting rod double piston rod, horizontal engines.

Much could be written on the present subject, but without practical value as a reference, unless the facts are demonstrated with illustrations. To define correctly what is considered the most eligible type is, to consult the evidences of those acquainted with many actual results; with this conviction the present subject will be diverted from for the present, and allusion next be made to Expansion Engines, commonly termed Compound.

COMPOUND OR EXPANSION ENGINES.

The remarks to the present have alluded in particular to single cylinders attached to one crank; now with this transmission of power, a low pressure of steam was generally used. Engineers after a time deemed it necessary, for the purpose of economy, to raise the pressure in order to reduce the consumption. This was of course correct, but the fact,—staring the projectors in the face,—that increase of pressure likewise affected the proportion of the materials, led to the adoption of two cylinders. From this issued the compound engine, or two cylinders of unequal areas acting on one, two, or more piston rods; and as may be presumed, many advocates and candidates stepped into the then new field of enterprise.

The economization of fuel is of such obvious commercial note in all localities, and very much so in our manufacturing districts where steam power is adopted, while our national and mercantile navies demand greater attention as to the reduction of the consumption of fuel. Now in order to increase the power evaporated from a given bulk of water *without increasing the space*, additional heating surface and pressure of the steam are imperative. The amount of evaporation requisite, depends of course on the pressure and consumption of the steam at each stroke of the piston; also the universal formula for determining the power of an engine is founded on two natural causes, pressure of the steam and speed of piston, but it must not be omitted that, momentum also assists the effect of an engine; inasmuch that, the greater the velocity, the centrifugal force increases in due proportion, and that bodies moving at high speeds must in all cases receive a momentum before the maximum movement can be attained.

The friction of the different classes of engines has from time to time been reverted

[illegible]

subject; but in all matters relating to science, an appeal to natural laws will always ensure an avoidance of superficial suppositions. To correctly determine the friction imposed by the passage of steam will be to consider the pressure, the bends and shapes of the orifices, because fluids or gases at given degrees of compression move in proportionate velocities, and the friction imposed will be due also to the form of the passages. The friction of fluids and gases passing through circular pipes is always less than through parallelograms or square passages, and the steam ports of an engine are [illegible] the steam during its transit from the boiler passes through a circular pipe, but on entering [illegible] differently shaped passage must be traversed. Now if the gas or fluid has to alter its form during its transit a certain amount of friction will be the result, in due proportion to the compression, shape, and [illegible] the form of the passages. Another fact also is certain: all moving particles while under pressure have a tendency to proceed in a direct line rather than a curved or angular direction. Steam or gas and fluids in this manner obey the same law, and [illegible] bends on the plane line and sudden ascends or descends from the same should be avoided. Compression or alteration of form is also an ally [illegible] passages [illegible] reduce the power [illegible] whether fluids or gases [illegible] the same, and [illegible] will be found more [illegible] square one. As for [illegible] double beat valves

with large casings for beam engines is a general practice, but certainly a better result would be attained with the present marine valve, and thus greatly reduce the valve gear, and connected details.

The number of the steam ports for that valve depends on the areas and length, and in the latest examples of marine engines we find a plurality of openings allowed for the admission of the steam, thereby producing economical results. The valve casings in all these examples are large in proportion to those of prior construction.

Next as to the friction of the steam in the ports, it has been proved to be inversely in due proportion to its velocity; therefore, the greater the pressure of the steam, the less the friction in proportion, always considering the same form of passages.

Having thus far alluded to the primary considerations relating to the present subject, attention will next be devoted to the most economical use of the steam, and the mechanical appliances for attaining the same.

The power given out by an engine is principally due to the area of the piston and pressure of the steam; the economical use of which depends on the amount consumed in proportion to the power exerted. Now looking at that simple natural law, it would seem that, to correctly design and construct an engine, would be to strictly observe the same. In practice, however, much has to be considered which in theory does not present itself; hence we find so many ideas on this important branch of science; consequently, since the commencement of the adoption of steam as an agent for power, there have been opposite opinions as to the better use of the same. One side advocates comparative low pressure with condensation, so that the power of the steam is aided by the effect of a vacuum. The antagonistic powers are devoted to the use of high pressure steam, discharging into the atmosphere. Now to define correctly the gain in each case, will be to remember the primary causes. When condensation is adopted, the temperature of the steam is of course sufficiently low to attain its normal state completely, at each stroke of the piston. The steam cylinders of those engines are larger than for the high pressure class, from the fact that power is due to the force exerted. Thus, a cylinder having 200 square inches area, with 80lbs. pressure of steam, = 16,000lbs., and equivalent in power to a cylinder 600 square inches in area, with only 10lbs. pressure of steam.

The presumed gain of the low pressure over the high engine is, that the former is assisted by a vacuum, while the latter is retarded rather than accelerated by the atmosphere; which is the theory held forth by the promoters of condensation, who hold that mode as perfection. And if that fact were taken as a guide, it would seem absurd to adopt any other class of engine; experience, however, teaches that "theory and practice," although naturally firm adherents, are often sadly separated by interlopers in human form.

The proportional figures just alluded to, clearly showed that pressure of steam greatly regulated the area of the cylinder; and "figures being stubborn truths," doubtless gave rise to the compound engine. Makers

[illegible]

[illegible] By this latter arrangement a further economy of space was gained at the expense of extra turning the smaller cylinder, and fitting the piston of the larger with an annular packing. This annular piston was not only introduced for stationary or fixed cylinders, but those of the oscillating kind were also made accordingly, which in their case may be truthfully termed "a sacrifice of economy of labour to preserve space." Indeed the excuse for our neglect of these curious arrangements is, that an allusion to the originators, their credit shall be in full force.

Having thus far described the relative position of the cylinders for vertical engines, allusion will now be made to those of the horizontal type. The necessity of placing the machinery for war vessels below the water line, led to the universal adoption of the arrangement now under notice, while for the mercantile navy also, it is fast gaining favour [illegible] and one of the chief causes [illegible] is the reduced height of the [illegible] engine room

to that of the vertical class, and thus allows overhead coal bunkers below the lower deck.

Enough has here been written to account for the attention devoted by engineers to the high and low pressure horizontal marine engine, of which the principal arrangements are as follows :—For single piston rod engines, the larger cylinder is further from the crank. The high pressure steam is admitted into two cylinders, secured in front of the larger, over and under the central or main piston rod. The pistons of the lesser cylinders are attached by separate piston rods to the piston of the larger cylinder, and thus a reciprocity of action ensues. The main piston rod is secured centrally of the whole, and transmits the power of the same to the crank shaft. Although three rods are in requisition, the principle of action and effect is as that for a single piston rod engine.

The next example has also three rods, but is an annular cylinder arrangement. The two side rods connect the larger piston and the central rod to that of the piston of the high pressure or central cylinder. It is almost needless to add, that the times of the action of the steam on the pistons are equal.

Now with the piston rod passing through the front cover, a certain space is required between the crank shaft and the former, but in order to reduce the distance, the trunk engine was introduced, and its application to the system now under notice has been in one case as follows :—The low pressure cylinder is nearest the crank shaft. The piston has a trunk at the front end, similar to ordinary single trunk engines. The high pressure cylinder is secured to the back end cover of the larger cylinder, while the pistons of each are connected by a single rod centrally secured, and the lesser cylinder is from its back end steam jacketed. The stuffing box of the trunk is prolonged and treated in like manner, by which means the cooling surface is greatly mitigated.

Ordinary single trunk engines have also been adopted for expansive purposes, by introducing the high pressure steam on the trunk side of the piston, and the low pressure steam on the opposite. Then as the larger area being operated on at a lesser pressure than that for the high, an equal force is exerted at each stroke of the piston.

The next arrangement to be noticed has the cylinders one above the other, the larger supporting the smaller. And in order to transmit the power of the steam correctly, the motions of the pistons are reverse in action; to attain which a perpendicular rocking beam is introduced, vibrating centrally, on bearings above the main piston rod. Now it is obvious that the higher pressure piston rod is connected to the upper extremity of the beam, and the lower end to the rod of the larger piston; by this means a reciprocal power and motion are maintained throughout, areas and pressures being of course duly considered.

The arrangements of the high and low pressure cylinders for return action engines are not much varied in relation to those lastly alluded to. In one example the larger cylinder is secured nearest the crank shaft, the lesser cylinder being secured at the back of the former. The connections of the two

pistons are by a single piston rod centrally secured in each. Motion is transmitted to the crank shaft by two piston rods secured to the large piston, and connected to the guide block in the ordinary manner.

Another arrangement consists of the high pressure cylinder imparting motion to one crank and the low pressure performing a similar duty to another crank; being in fact, in principle of action, as for ordinary engines, with a steam pipe communicating with each cylinder. But the connection with the condenser in this case is from one cylinder only. Now in order to reverse the relative motions of the pistons with this class of engine, the low pressure cylinders are secured on the stringer plates, and the high pressure cylinders are fixed on the upper side of the low.

Beyond the piston rod guide channel is suspended a perpendicular beam, one extremity of which is connected to the crosshead, and the other or upper end is connected by a rod to the lesser piston rod crosshead; by which connection a reverse action of the pistons takes place.

The speeds of the high and low pressure pistons in each case, be it remembered, are alike in all the examples heretofore alluded to. Now small cylinders, particularly with high pressure steam, will allow a greater velocity of piston with more economy and safety than with large cylinders. And the most practical means consists of the introduction of spur gearing, situated centrally across between the low pressure cylinders. The nigh pressure cylinders are secured on the former, opposite the gearing, the spur wheel of which is of a much less diameter than that on the crank shaft. Now the ratio of these wheels will depend on the speeds of the pistons and pressures of the steam in proportion to the grade of expansion.

The introduction of spur gearing to increase the speed of the crank shaft, has been long ago discarded. But there is no reason, however, against its adoption for regulating the diversity of the speeds of the pistons of cylinders under different pressures, which is finally common to one shaft.

The principal gain in the adoption of compound engines consists in the arrangement for the use of the temperature of the steam. The higher pressure passes into the smaller cylinder at a given temperature, and after exerting its required duty, enters the larger cylinder. Now it will be understood, that each temperature should be preserved, *per se*, *i.e.*, the pressure of the steam in the small cylinder is not reduced by radiation, but rather by the effect of expansion; and the action of the steam in the larger cylinder will, of course, be likewise affected. Makers and advocates of compound engines hold this argument, however, only as a part of the gain to be derived by their adoption, because steam acting on a piston of course exerts a known force; and that the piston and connecting rods, with the crank shaft of any kind of engine, can only be proportioned from the area of the piston and pressure of steam; and with this natural cause and effect, as a rule, the devotees to the compound arrangement claim due attention. Independently of those facts as matters of importance, there is another gain to be obtained, which is uniformity of the power exerted, or rather produced. But this of course is also due to the proportionate pres-

sures and areas, and engines of the class now under notice will always retain the advantages now set forth.

Compound engines proper are actually two cylinders in the place of one, to produce the same effect, because the advantage of expansion is simply due to pressure of the steam, and the consumption of the same for a given time. A cylinder of a given diameter can, of course, only admit a certain quantity proportionately. And by increasing the cubic contents, or, rather, the area of the cylinder for the same length of stroke of the piston, a greater amount of steam can be introduced, but not necessarily; while a uniform action greatly lessens the shock produced by the lead of the valve; and final reduction of the pressure of the steam will, of course, affect the temperature, and thus cause a more ready and perfect condensation: therefore it may doubtless be surprising to some people that compound engines are not more universal.

Expense of financial outlay is the principal objection, while at the same time, the results yet attained are not seductive enough to renounce the ordinary mode of working expansively; but in order to preserve a correct motion, and dispense with the compound type, three cylinders are sometimes introduced, a corresponding number of cranks being imperative; while in some instances six cylinders are adopted, but the same number of cranks retained, but the increase of the total area of the piston greatly affects the torsion of the crank shaft; while also an objection can be raised against the necessary increased weight of materials, over the ordinary engines.

As for the pressure of the steam being conducive of proportionate concussion due to "lead," there is not the least doubt that there is some cause for the prevention of the same; but this evil is, however, in most cases, greatly exaggerated; for it must be remembered that the steam enters the cylinder at a given pressure, and continues its flow till the cut-off ensues.—Expansion now performs its duty.— At the last part of the action of the steam the temperature will be the lowest, and the power, of course, the least.—Now, were it not for the impetus given to the piston at the commencement of the stroke; at the point of exhaustion, its speed would be sensibly lessened; and in practice it is found that this inequality of motion is not depreciating, and still less so, particularly, with high velocities.

The most economical use of the steam should ever engross the attention of the Marine Engineer; and it would not be out of place if the actual results attained were a little more public. The main points to be considered in all types of engines are: attainment of the utmost power at the least cost and continual expenditure; because, as before stated, the commercial question is always the foremost in all enterprises, of all classes and arrangements. The wear and tear of an engine, due to its type and construction, should ever meet with due consideration; for what is pleasing to the eye may not always be the best for financial interests; while at the same time it must not be forgotten that good design should be strictly retained in a practical form.

HEAT AND COMBUSTION.

The cause and effect of heat is a subject worthy of consideration: perhaps it is difficult to point out any branch of science of more actual importance, inasmuch that the propeller may be correctly proportioned, and the engine the most simple; but the effect of both depends on the evaporative powers of the boiler.

Now the *cause* of latent heat, undoubtedly, emanates from nature; but there are two modes of producing temperatures above the ordinary degrees—combustion and friction. The former may be said to be a natural production, while the latter is an artificial attainment, commonly known as electric heat. The rays of the sun, too, acting on the waters of the earth, impart a truthful lesson worthy of attention from all. When it is remembered that a perfect evaporation is in constant action, in proportion to the required supply for vegetation, it cannot but be acknowledged that "Nature is the foundation of Science." When heat is caused by friction it is generally unintentional with the engineer; in fact, with mechanical operations, it is well known that this attainment is looked on with dread rather than with desire.

The action of combustion will—due to natural laws—in all cases, produce heat; and the higher the temperature attained the more perfect is the required effect; but before entering further into the chemical and practical questions of combustion, attention will be given to the *effect* of heat, which as the action of light will penetrate with a mysterious power into the most minute crevice or opening; also forces through bodies of immense substances, the exterior of which throws off a vapour of a given temperature; and the surrounding air becomes rarified by coming in contact with the heated body, and thus affects the whole. Now, as to the velocity of the penetration of heat, two essentialities have to be considered: the temperature and conductibility of the body as an accelerator. Therefore the penetration of heat through a given substance will, of course, occupy a given period; but it is not essential, however, to double the temperature to retain the velocity in proportion. This effect is due to the state of the caloric at given degrees; it is well known that the higher the temperature of any gas or vapour, the less its density. The friction of fluids and gases are likewise affected, and the greater the state of compression, the lesser the friction during transit.

The effect of heat in marine boilers is to produce evaporation, and with what economy is the main point of consideration, so much so that there seems to be many doubts as to the correct thickness of the flame plates of a boiler for given pressures. Some authorities advocate thick plates, while others recommend those of a lesser substance. Now to consider which will be the most effective, will be to remember the action of the flame and heated gases.

The time of conduction of any material is not a matter of universal consideration, because it is too often forgotten that, the nature of the appliance is the chief portion worthy of attention when practical results are required; and that the velocity of heat will in all cases depend on the conductive powers

of the material operated on, and the bulk of fluid beyond the same.

A fire box composed of thick plates, and a given depth of water surrounding the same, will produce a certain amount of evaporation. Now to increase this effect will be either to reduce the thickness of the plates, or increase the area exposed to the flame, and to define the most practicable mode of attaining the better result, two essentialities must be noticed, pressure of the steam required, and the area of the grate surface. The strength of all materials decreases in due proportion to the increase of the temperature of the same, *i.e.*, a thick plate with sudden evaporation and shallow water is not equal in durability of strength to a thinner plate with a greater amount of water surrounding the fire box; therefore, the main point, or rather the fact to arrive at is, the correct proportions of the requisitions in being. It may be argued that, a thick plate being of greater substance than a thinner, must naturally be the stronger; but the fact that the conduction, or rather the penetration of the heat, regulates the strength of the material operated on, disperses entirely the law of the cold tenacity of metal.

If the temperature is not the same throughout the thickness of the plate, that portion most operated on by the flame will be reduced to the *natural* substance, and truthful and practical evidence of this can be seen with the laps of plates, heads of stay bolts, nuts of the same, corners of angle iron, &c.

Another fact to be noticed is the durability of a worn plate to that of a thicker one under the same exposures or effects of the flame; which, when acting on the surface of the metal, disturbs the fibre of the same, and until the process of carbonizing can be said to have been partially completed, the destructive properties of the gases are in continual operation.

Next with reference to the amount or depth of fluid surrounding the plate. Much depends on the evaporative properties of the latter, in a chemical point of consideration, because, in all phases of science, knowledge of the origin or natural state of the cause and effect of the portion investigated, will give a truthful result. Likewise, the natural adherent faculties of all materials are worthy of the greatest consideration, also the alteration of the same while under the artificial process of manufacture.

The ore, for example, in its state of purity, is a composition of certain minerals; the cupola and blast separate the same, and metal in its primary grade is the result; the next stage is purification, or the production of adhesion and elasticity in proportion to strength; the third operation is the particular manufacture for practical utility, and the completion: the application of the material. This is the actual category of the boiler plate, and to correctly apply the same, is to understand how it is produced. To the present, iron has only been alluded to as the material available for marine boilers; there are, however, other metals, equally, if not better conductors of heat, and more durable than that already mentioned.

The resistible nature of all metals consists of three kinds, torsion, tenacity or cohesion, and compression or crushing strain. From those three natural causes are produced many

effects. With torsion, a distorting tensile strain may be said to be resisted. With a tenacious nature, ductility of material is available, as with cohesion; but with compression or crushing, a total disturbance of the fibres results. But the elasticity of any material must not be confused with the ductile properties of the same. Both natural effects do not emanate from the same cause; as for example, india-rubber is certainly the most elastic of all solids, but ductility is its least property for practical purposes.

The metals most utile for conductors of temperature, are silver and copper. The former, from its scarcity and non-resistance to tensile strains practically, is more ornamental than useful. The latter metal is better adapted for practical purposes, being stronger and more plentiful.

Ductility of metal, in the practical sense of the term, means a suitable resistance to tensile strains while being altered or extended in shape uniformly. Cohesion is another faculty which all ductile materials possess, hence copper may be truthfully said to be the best conductor of heat for practical purposes. And for fire boxes, it has been found long ago to supersede all other metals, not only from its powers as a conductor, but also from its natural availability for manufacture, without jointing, and this advantage can be readily appreciated when it is remembered that all joints of plates are the weak parts, and bad conductors at the same time.

Heat penetrates through bodies disproportionately, due of course to their fibre and nature, and such as deal and flannel or felt, may be said to be the retainers of temperature; for while flannel will preserve ice, the same effect is gained by a similar clothing for steam pipes; and the lagging of boilers is now deemed one of the best means to economize the heat.

Thus leaving the subject of heat for the present, attention will now be briefly devoted to combustion. The natural transit of all gases is to ascend, hence the effect, that the crowns of fire boxes, upper surface of tubes, top side of tube plates, are considered better evaporators than the remaining portions of the whole; while in all cases, flame will be proportionately effective according to the draught attainable. At the same time, this artificial propagator can be converted into a destructive agent; for example, the ordinary blow pipe and the cupola blast pipe, are alike in principle and effect. All gases, as before stated, naturally ascend, hence it is readily proved that the space above the fire bars should always be sufficient; also, for stoking, high furnaces are advantageous.

This portion of the many duties requisite for steam navigation, is, perhaps, the least generally considered by those not *actually* acquainted with marine boilers. The theoretical rules laid down for correct stoking are often impracticable: "keep a clean fire," is the mental order. Now we find in practice, with bad coal, this mandate to be almost a *cruel joke*, because coal contains iron, more or less, and thus clinkers or slag are profuse: therefore, to "keep a clear fire," under those circumstances, would require to be always agitating the same. Patents innumerable have been granted for this purpose, but to the present none have been productive of the

requisition conducive to economy. Coal, in its natural state, is a combination of solid matters, from which emanates certain gases; and the time allotted for combustion can only be regulated by existing circumstances; therefore, to lay down a rule for stoking, is similar to an individual mentally making out the programme of a route before travelling through an unknown land. It is not here argued that science is helpless in any case, but rather, that theories made in the study are not generally practised at sea, and particularly those relative to combustion, or rather the often proposed mechanical means to accelerate the same.

The correct mode of cleansing a fire is, to carefully remove half the upper portion of the fuel to the opposite side, then remove the slag, and reverse the operation to clean the whole. This can be carried out in the full sense of the meaning with wide fire grates, but with narrow furnaces, the rule is far from absolute.

Presuming next that the full effect of the fuel is operating in the fire box, it will be well to follow metaphorically the duty of the flame throughout; beginning with the fact that the action of the flame as it passes the bridge, is to rush into the combustion chamber.

This last mentioned cell should be amply large to allow for the accumulation of the gases; in fact, it is actually the receiving and the purifying cells at the same time; and there is not the least doubt that the action of combustion is greatly accelerated by the gases amalgamating before entering the tubes. Besides, the adoption of a small chamber, beyond the fire grate, is perhaps the best example of the ignorance of combustion that can be conceived; but, to the credit of Engineers generally, such an abortion is a rare occurrence.

Presuming a sufficient allusion to an ample combustion chamber, and the acquisition of the same to have been duly explained, the further progress of the flame will now be followed.

The next portion exposed to the action of the flame is the tube plate; it will be well to observe in passing, that much attention is due to this detail of the marine—tubular—boiler. The plate in question is a perforated portion, or a semi-solid preventitude of direct action —of the flame and gases—from the combustion chamber. Many authorities disagree as to the proportion of the evaporating properties of the plate in question. Some advocates for economy, rail against the advantage of the flame acting between the ends of the tubes. Other sages—worthy of attention—energetically do their best to impress on the minds of Marine Engineers and shipowners, that the supporting plate of the tubes is the main agent for evaporation. Now, to entertain *one* opinion only, will be unfair and prejudicial, to say the least of it; hence a compilation of the remarks of a recognised author will of course be admissible. The late Charles Wye Williams, Esq., Assoc. I. C. E., &c., in his work on "Steam Generative Power," states thus—on the assumed value of tube surface:—

"Against the insufficiency of the tubes as heat trans-
"mitters and steam generators we have the hitherto
"neglected surface of the *face plate*—presenting a face to
"the direct action of the hot currents from the furnace.
"Assuming that the orifices of the tubes occupy one-
"fourth of the gross area of the face-plate, the practical

"heat-transmitting portion will be the remaining three-"fourths.

"In general terms, then, we may say that whatever "may be the gross area of the face-plate in marine boilers, "three-fourths, certainly not less than two-thirds of it, at "the least, should be capable of receiving the direct action "of the heated current from the furnace."

* * * * *

"Hitherto the face-plate has been regarded as a mere "mechanical contrivance by which the tubes were held "in their places for preserving certain distances between "them to enable the water to surround them. Strange to "say, no idea whatever appears to have been entertained "that the face-plates had any heat-transmitting or steam-"generative property *of their own.*

"As perforated diaphragms, however, which they prac-"tically are, their true function and merit consists in their "suddenly intercepting the current of hot gaseous pro-"ducts from the furnace, and appropriating a portion of "the heat of that current to the generation of steam in "the water immediately behind, and in contact with "them. The amount of heat transmitted to the water "through them will then be the measure of the steam-"generative value of these plates.

"The now proposed system, as I have elsewhere stated, "consists in substituting for the ordinary long tubes "employed, sets or series of short tubes or flues at short "distances apart, the ends of each set or series being fitted "into tube or face-plates like those into which the long "tubes are united."

On the "law for regulating the application of the plate surface:"—

"Without reference to the tubes, it appears by practice, "that one square foot of the available area of the *face-"plate* is required for each square foot of *grate-bar-surface*, "and this may safely be taken as the basis of a law of "proportions for future calculation. Thus, for a furnace "of twelve square feet, the heat-transmitting area may "be found in a single face-plate of the same extent.

"But, take the case of a large steamer like the *Warrior*. "Here each furnace, say of 6 feet by 3 feet, will have 18 "square feet of bar-surface. The gross area of the face-"plate of each stack of tubes may also be taken as equal "to 12 square feet—one-fourth of which being occupied "by the open ends of one hundred tubes, there will remain "but 9 square feet of heat-transmitting surface, against "18 square feet of grate surface—just one half of the "required area for evaporation. In such case a *second*, "or even *third face-plate* will absolutely be required for "the transmission of the heat from so large a grate. If "this quantity be not supplied, one-half the generated "heat must, of necessity, pass away as *waste* by the "chimney. Estimating the evaporative value of a second "face-plate at two-thirds that of the first, and of a third "at one-third that of the first.

"It is the presence of this large quantity of heat in the "chimney stack that has set the ingenuity of engineers at "work in considering how it might be utilized. Among "the modes of effecting this object, the process for heating "the feed water, and also the *super-heating* process, was "invented. Nor can we be surprised at the high tem-"perature of the escaping unappropriated products, seeing "how little really heat-transmitting surface has been "provided."

Mr. C. Wye Williams seems to have been rather self-confident as to the valueless surface of the tubes in proportion to the plate, *per se.* The talented author, no doubt, is partially correct in his theory that the surface of the tube-plate is worthy of consideration, but he is certainly surprisingly severe, when he states, "strange to say, no idea whatever appears to have been entertained that the face-plates had any heat-transmitting or steam-generative property *of their own.*" The author's work is a well digested production, and worthy of a better cause, *i.e.*, as far as regards denouncing the recognised and well proven value of tubular surface as the main surface for evaporation. He not only condemns the tube surface as of little or no value, but he also appears to accuse the powers that be, of negligence, thus:—

"In truth, the prevailing opinion and practice comes to "this, that, where larger engines are required, and greater "steam power is to be supplied, engineers have no alter-"native but that of enlarging the furnaces and increasing "their number to a mischievous and wasteful extent, "together with an addition to the length and number of

"the tubes, under the erroneous impression that a corres-"ponding increase of evaporation would be the result. "Both of these changes, however, are alike destructive of "economy and efficiency.

"Had they only examined (as should always be done) "the action of the several parts of the boiler,—its several "organs, so to speak,—when each could be seen and its "merits appreciated, the perpetuation of the great mis-"takes we are daily committing would have been avoided, "and the true heating surface discovered. Yet this neglect "in examining the action of the several parts or organs of "the boiler is not less culpable than if surgeons were to "ignore dissection, or the examination of the functions of "each organ of the body, and trust to observation of the "exterior human frame.

"In a word, the *insufficiency* of the tubes, as steam "generators, has been practically their real *protection*. "Had they transmitted heat sufficient to generate any "quantity of steam, they would soon have been destroyed, "as no water could then possibly have found its way "between them, and where nothing but steam from below "would have existed.

"How lamentably erroneous, then, has been the whole "system of tubular boilers, and the reference to the tube "surface as the measure of the evaporative power of a "boiler. The sooner that this fallacy is exposed the "better, otherwise we must go on floundering in a mis-"chievous and expensive course of error."

According to this theory, if correct, the sooner we return to the old flue system the better. Before, however, resigning (*sic*) what has been gained since the abolition of the flue action, it will be well to define the present system universally carried out.

It is known that, by the action of the flame passing through perforations, a certain amount of retention of heat results, and from the distribution, or rather forced separation of the flame and heated gases, in the combustion chamber, a thorough amalgamation occurs, which can only be maintained by the tube or contracted flue system. According to the theory held forth by the author now quoted, the tube-plate is the vital generator of the whole structure available for evaporation. The advocator not only asserts this, but in his able work he attempts a practical demonstration from experiments. In some instances three combustion chambers were introduced, thus producing startling results, for the author states :—

"Here there was a practical illustration of the value, "not only of *one* but of *two* face-plates, in addition to "the original one belonging to all tubular boilers. Yet "the value of the face-plate, as a heat transmitter and "evaporator, has hitherto remained unappreciated and "even unnoticed."

Now, with all due deference to an able authority, it cannot be maintained for a moment that the present system is totally wrong either in theory or practice; for what is now considered correct, is that, the flame and gases being contracted and confined for a time in a series of tubes of a *lesser substance* than the *tube-plate*—or any other portion of the boiler—a certain increase of evaporation must be the result.

The tubes, let it be remembered, are supported at each end by two separate plates, the effect of these latter being the *star* of the quoted author's vision.

Now, on consideration, it will be understood that the back end plate—or that nearest the fire—receives the flame but partially; while the outer, or front plate, loses its effect entirely, and can in no case be accounted as an evaporator. With this as a truth, and practice to prove the same, it is difficult to account for the imagination of a really practical man straying, as it seems to have done, according to the remarks he published.

To return to the action of the flame in the combustion chamber, and its exit from the

same. The tubes of almost all modern marine boilers have a rake, or an inclination from the level at the front end. The object of this is to ensure a better draught and evaporation. Now the tube-plate receives some of the heat of the flame, while the tubes retain the effect of the remainder.

Next as to the length of the tubes. To the present, Engineers have decided, almost invariably, on a length of from five to seven feet, and two-and-a-half to three inches in diameter. In some cases the length is increased to eight feet six inches, but the hull space allotted is the principal guide.

When calculating the evaporative powers of the tube surface, the entire circumference is often considered alike as an agent of, or rather a conductor of heat. Now, if the law of evaporation and free transit of all gases be remembered, it will be noticed that the tendency of the same is to ascend; hence in the case of the flame in the tubes, the upper part of the latter is mostly operated on, and the better evaporative portion. Full evidence of this is seen in all cases where the flame is contracted, such as the crown of fire boxes, upper plates of combustion chambers, &c.

To ensure the inner portion of the tube to be entirely filled with the flame, tubes of a conical shape might be used advantageously, but the greatest barrier is, the withdrawing of the same from the front end without the removal of the plate.

Suppose now the flame to have passed through the tubes and is in the smoke box—which receives the smoke and worn out flame from the tubes, hence its name.

In practice, however, it is often found that much of the effective caloric finds its way beyond the compartment in question. This loss of effect is due to two causes—incorrect proportion of the combustion chamber, and malconstruction of the smoke box; and as an example, the experiments of Mr. C. Wye Williams may now again be adverted to as an example.

He states that by reducing the lengths of the tubes and introducing intermediate combustion chambers, "or heat chambers," much gain in evaporation was the result. Now this was not due—as he states—to the "face" or tube-plates, but rather to the time allowed for the second and third amalgamation of the gases.

It is not erroneous to conclude that a series of combustive explosions ensued within each combustion chamber, and thus the rarefication of the flame and gases within the *tubes* increased the evaporative properties of the latter. It can readily be admitted that, the front tube or "face" plate is productive of a certain amount of evaporative power; but as the main agent, and superior to any other portion, practice contradicts; while, for cleansing and repairs, with an intermediate combustion chamber, by what means could the latter be internally accessible? truly, only by a door on the crown, which must be removed before even an examination is available, to say nothing of the necessary position of the stays as an inconvenience.

Engineers have for some time seen the folly of a maze-like combustion chamber in the form of flues, and thus it is hardly to be presumed that to return to the same will be the present or future practice.

The action of the flame in the smoke box is neutral as an evaporator for some portion, but lately water spaces have been introduced above and between the doors, and thus extract some of the heat during its ascent in the form of flame and smoke to the chimney, after which it returns to its origin.

Truly here is a wonderful demonstration of the *truth* of natural laws, "nothing is lost"—metals may be worn with friction, slow in operation but certain in effect—wood may decay—fuel may be consumed; but all are preserved by nature, and a change of form only ensues, because cubical contents or bulk of matter is not destroyed.

HIGH AND LOW BOILERS.

The proposition of any commercial scheme or mechanical contrivance is, perhaps, within the province of many; but the actual working of the scheme is devolved on the few. The preceding remarks on combustion are the primary descriptive matter of the theory, while the present will treat of the practice.

Marine boilers at the commencement of their use, were not noted for their simplicity of arrangement, but rather for the internal peculiarity of form. The designers of that age seemed to look on flues as the only means of distributing the flame, and those heating spaces were fair representatives as to what could be done in the way of intricacy combined with practice. Perpendicular flues connected those of horizontal position, and in some instances angular and inverted passages were introduced, and a section—of some of the examples—in plan, imparted a fair idea as to the design of a maze.

The cause for cleaning is one of the evils of combustion, as well as other operations, and thus the arrangement in question was not perfection. It was soon clear that where the flame could traverse, a man or youth could not pass, thus much space more was requisite for the purpose of cleaning than for combustion; therefore, the labyrinth arrangement was found to be faulty from the fact alluded to, from which, doubtless, emanated the now universal tubular boiler.

The high boiler of the present day is a case or shell containing a series of fire boxes situated near the bottom. The combustion chamber is either in one compartment, or forms a separate portion connected with each fire box. The tubes are secured above the fire boxes on a line with the same. The tube-plates form the side of the combustion chamber, and that of the smoke box, the latter being at the front of the boiler connected to the uptake that is prolonged to the top of the shell, to which the funnel is secured. In some cases the uptake of each boiler forms a breeches piece, the funnel being connected to the single or upper portion. Now as to the height of the boiler above the tubes, much depends on the depth of the hull of the vessel; it may be added, in passing, that the more steam room available the better.

The exterior shape of the shell depends greatly on the arrangement of the interior parts of the hull, it being remembered that the latter receives the former.

The general form is flat, top and bottom, connected with bold curves; in some cases, a portion of the top is raised, longitudinally throughout, to allow more steam space. This

is a common practice for small boilers, and for those of larger power also where height is available. The shape of the ends or sides is due to the beam of the vessel and the length of the fire grate. The bottom portion near the back end is generally raised angularly to connect with the same, it being more available for evaporation when raised, than when flat, or on a level with the bottom of the fire box. The front portion of the shell—above the level of the tubes—generally projects, both for increase of water and steam spaces. The bottoms of the fire boxes are generally inside the shell; it must be added, however, in some instances, the bottoms of the boxes and shell are on the same level, in fact, the base of the former commencing at the bridge; and this arrangement reduces the height of the boiler, minus the water space, a matter of great importance sometimes—it will thus be understood that the word "high," is only in this case a technical term.

The next class of boilers is the low kind, and their origin may be traced from the following causes:—When shallow rivers have to be navigated, a distribution of weight superficially is imperative, and when a clear deck space is requisite, any raised portion reduces the area accordingly. For war purposes, also, it is necessary that the machinery should be below the water line, to prevent disablement from shot or shell, because a steam vessel, with non-effective machinery, becomes less navigable than the once famous sailing crafts, and thus protection is essential.

The arrangements of the class of boilers now under notice, are perhaps more varied than those of any other type. This may be said to be due to the requisition of increase of power with a reduced height and breadth. The exterior forms of the shells of the boilers in question are cylindrical, flat, and angular sides, all in requisition for certain arrangements and localities.

With the cylindrical shell, the internal arrangement in some cases is thus: the fire box—either in one, two, or more compartments—extends direct from the front of the boiler to the combustion chamber, the latter and the water space completing the entire length of the boiler. The tubes are dispersed at the sides and top of the fire boxes, passing to the front of the boiler, unto which is connected the smoke box. By this it will be understood that, the commencement of combustion and the termination of its use, are at the same end of the boiler, or similar in arrangement to the ordinary high boilers already described.

The next arrangement worthy of comment, is that of the direct acting kind, *i.e.*, where the fire box, combustion chamber, tubes, and smoke box are in a direct line with each other, as that of a locomotive boiler. The internal arrangements of the flat and angular shells are much alike, the shape at the sides being suited to the beam of the vessel. The other examples most universal are as follows:—The fire boxes are at the end of the boiler nearest the engines. The combustion chamber is in front of the fire grate for the entire width of the fire boxes, and extending at the outer side—or that nearest the hull—for the entire length of the shell. The tubes are secured at right angles with the fire boxes beyond the

same. The smoke box is in a line with the extension of the combustion chamber. With this arrangement of details a short tube is imperative, while the requisite number makes good the surface required.

The next example consists of the fire boxes being situated at each end of the shell, with separate combustion chambers and water spaces, thus forming the entire length. The tubes are secured at the side of each fire box, having a return action, and height being of importance in the present case, the smoke from one end of the boiler—fore or aft—returns through a separate compartment or box extending from end to end of the boiler; for, by this arrangement, the products of combustion can escape through one funnel.

In order to obviate the necessity of the additional smoke box—as in the last example—the following arrangement has been devised: the fire boxes are centrally situated at each end of the shell; the combustion chambers are entirely separate, extending partially across the shell; the tubes are at both sides of the fire boxes. At each end of the boiler—at the outer sides of the fire boxes—are second combustion chambers; a second series of tubes from these connect to a smoke box, centrally secured in the length and breadth of the shell. By this it will be understood a double return tubular action is attained, and it may be added in passing, that the entire length and breadth of the shell in question will not be increased in proportion to those of the preceding examples.

In all devices for arrangement of boilers, a similarity—or, rather, an equality—of flame-action should be maintained, which in the last example is certain. The types of boilers now under notice are worthy of more attention than has been the universal practice. This may be said to be due to the isolated requirement of the case; but should a future improvement, for national purposes, be desired, the British Marine Engineer will as before now be equal to the task.

END OF PREFATORIAL REMARKS.

CHAPTER II.

GENERAL ARRANGEMENTS OF ENGINES.

It is difficult to point out any portion of the general duties of an engineer of more importance than the classification of the machinery requisite for the hull of a vessel; indeed, it can be truthfully maintained that a more arduous task cannot be set before the professional faculty. Yet, with this as a certainty, how little is the appreciation universal! How seldom is considered what is due to those, who, with untiring energy, consume health, time, and money, to attain a given effect of mechanism!

The general spectators, gazing on the complicated arrangement of the component parts of a pair of marine engines, are merely mechanical observers. The position of a given lever or shaft—right or wrong—is all the same to them; doubtless, the thought rarely, if ever, arises that skill and fertility of the brain must have been well tested to produce the subject before them. Singular enough, this is the case with all crafts and professions: that portion seeming the least effective to the non-conversant has often been the most difficult to attain by the producer.

For example:—"the expansion gear of a marine engine;" to the outside world it looks pretty, but as to the care and thought it has often cost, that is the last matter considered. This coldness of observation is mainly due to ignorance; because the effect, that should be, passes away harmless, unseen, unfelt, and, therefore, unknown, to the unconscious spectator. As with engineering, so with chemistry, mathematics, and other of the higher branches of science. The engineer, however, has this advantage over most professional representatives: all science must be at his command more or less, according to the requisitions; and thus a general knowledge is acquired.

John Fowler, Esq., M.I.C E. (late president of the Institution of Civil Engineers), in his well-digested inaugural address to the members, January 9th, 1866, duly acknowledges the advantages of a practical knowledge:—

"The mechanical engineer deals with the most varied "and numerous subjects of all the branches of engineering. * * The mechanical engineer, generally, also "executes the designs of the gas engineer. * * "Allied with the mechanical engineer is the naval archi-"tect; and only a mechanical engineer could have con-"structed the vast steam ships of modern days."

How often it is the case that the advantages obtainable in science are neglected because those who are ignorant of its value are even doubtful and thus timid to undertake the responsibility offered. Again, to exemplify, too occasional is the contrary fact, that many per-

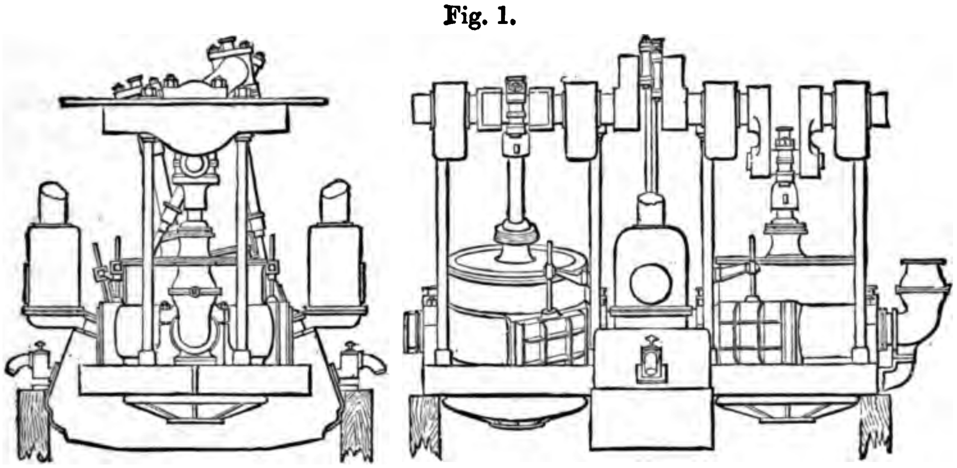
sons become victims to their mechanical mode of grasping difficulties, which results in a total failure: they then blame science, and forget that seeds sown by nature, even on good ground, require cultivation.

In another sphere a large amount of surprise is often displayed by the friends of a student of engineering, that another labourer in the same field should surpass their hero in knowledge, whose influence, wealth, and position are vastly superior to those of the more fortunate student. The true cause for this effect is, that the student, with his influential friends to back him, is generally pushed, rather than plodding on, as the poorer, but better man; while, perhaps, he is too conceited to know how ignorant he really is.

It is not every one that *can* be an engineer. Three essentialities are imperative: sound education, quick perception, and a decided taste for conception, or originality. The only reason why professions are above trades is, that the members of the former work mentally, but those of the latter do it mechanically—with exceptions of course in either case.

Fertility of brain power is a boon to all mankind, but its possessors are few proportionately to the many; nevertheless, much can be done by self-improvement: because thought produces originality of ideas, and the mind expands with a repetition of action. Rarely is it the case that an inventor stops at one point; indeed, it has been said by some medical authorities, that the possessor of an inventive mind is its slave. Theoretically, this may be correct, but in practice the intelligent mind derives such pleasure in its researches that it is invigorated rather than weakened by constant action.

Fig. 1.



MODERN OSCILLATING ENGINES BY MESSRS. PENN, RAVENHILL, ETC.

OSCILLATING ENGINES, FOR PADDLE-WHEEL PROPULSION.

The arrangement of any pair of engines will depend on the class, size, position, and mode of condensation, which in some cases permit a constant arrangement; so that the oscillating engine is now universally used for the paddle wheel; and the arrangement, as illustrated above, is by Messrs. J. Penn and Son, who are said to have been the first to introduce those engines to any extent; after them came other firms — Messrs. Rennie; Messrs. Maudslay, Sons. and Field; Messrs.

Ravenhill, Hodgson, and Co.; also Messrs. R. Napier and Sons, of Glasgow, and other notable makers—adopted the same with good result and credit.

The working mean position of the cylinder has not in all cases been perpendicular, for in some instances an angular position has been deemed preferable; this latter may be said to be available for a long stroke of piston and shallow hull. Most, if not all the firms, of the present day, prefer a vertical mean action, on account of the weight of the cylinder and appendages straining the piston rods, &c., less.

The modes of attaching, or rather suspending, the cylinders are much alike by all the firms. The trunnions are turned and bored in the ordinary manner, while some makers prefer to cast the trunnions separate, and secure them by studs. The stuffing box is often a separate portion, secured to the condenser, a projection being prolonged from each trunnion for the purpose of making the requisite joint.

The distance between the centres of the cylinders are subject to the air pump and condenser. With small examples, or those below 50-horse power, collectively, the air pumps have a vertical action direct, under the intermediate shaft. The condenser surrounds the air pump, and is the most compact form that can be made, due, of course, to the annular arrangement; while, for engines of larger power, the pump can be increased in due proportion, and the beam of the vessel need not be affected in the same ratio.

The centres of the bearings of all kinds of engines should be as near the cranks as possible; hence, with the class now under notice, the length of the intermediate shaft is affected by the arrangement of the condenser. In order to retain the minimum distance between the inner bearings of the shafting, the air pumps are situated outside the condenser, at an angle with the centre of the shaft.

The top frame, or entablature, is often in two portions, connected transversely. The section of the portion on each side of the shaft-bearing is generally that of a parallelogram. The top portion is flat, generally on a line with the centre of the shaft. In other examples, the framing recedes at an angle from the centre line; others again are angular in shape, while a fourth is curved, both above and below the connection, with the beams of the hull.

The columns for supporting the entablature are not, in most examples, situated centrally of the shaft bearings; and the general cause of this is, the space required for the removal of the cylinder, and also to dispense with additional guide standards for the sliding quadrant of the valve motion.

For engines of large size, or above 100-horse power nominal "cross frames" are sometimes introduced outside the wing columns, these frames act both as a stay and distance piece between the entablature and lower frame.

The universal mode of imparting motion to the pistons of the air pump is usually by a crank, while, in some cases, eccentrics have been adopted; but preference is generally given to the former, particularly with long strokes, of about three feet six inches. The connection of the piston to the crank pin is

usually by an ordinary connecting rod. The piston rod pin is sometimes guided by side guides, while, latterly, those have given place to trunks, which are more simple, with a better result.

The condenser is situated between the cylinders, and prolonged underneath the angular air pumps; but when the latter are vertical, the condenser is on each side of the pumps.

The angular air pump is the most universal; also the crank action as a prime mover. Two pumps are more general than one, on account of the reduction of the diameter of the same. The injection water enters the condenser midway of the trunnions, so that immediate condensation is attained.

The positions of the suction and delivery valves are mostly alike by the several makers. The suction valves are directly under the piston; which is also fitted with a similar set. The discharge valves are arranged within a separate compartment at the top side of the barrel. The pumps for the condensers are mostly single acting.

The lower framing of the engines is generally a plain section of the letter L shape, front and back of the cylinder; and projections are cast on the sides for securing the trunnion plummer blocks, &c.

The motion requisite for the feed and bilge plungers is generally derived from the cylinder, while in other cases separate eccentrics are used.

The steam cylinders and their appendages are in combination with the starting gear and valve-motion: and of those details we will next notice that used for the purpose of balancing each cylinder; two slide valves are mostly introduced, placed in duplicate positions; and the gear for imparting the motion is that of the eccentric kind, with a sliding quadrant, to counteract the oscillation of the cylinder. The attainments of starting, stopping, and reversing the engines are produced by the ordinary link-motion, or the loop end eccentric rod; and counterbalance on the crank shaft.

Much controversy has arisen amongst some authorities as to the better kind. One party recommends the link-motion, while the opposite side prefers to use the single eccentric and loop end to the rod, as the better for disengaging. The fact actually resolves itself into this; with the link-motion and two eccentrics; no actual disengaging of the connection of the eccentric with the sliding quadrant is requisite. Now, with the single eccentric and looped rod, an entire disconnection must ensue, also the valves must be moved independently to attain the desired effect.

For small river steamers, where sudden and certain manipulation is necessary, the single eccentric and counter balance is perhaps the better; because, as the slide valves of large engines too often require great power to shift them—when under steam—at starting, in their case the link motion is the most efficient; but in any case the starting gears for oscillating engines, depend much on the power of the engines and balancing of the valves.

The single eccentric arrangement is thus: the eccentric rod, at the lower end, has a provision for the pin of the working gear sweep, or sliding quadrant, as some term it. The loop on the rod prevents the same from

leaving the line of action; the constant contact of the rod and pin is attained by a spiral spring, casing, &c. The disconnection is caused by a lever pressing on the sweep pin, and thus raising the rod from its connection. The means for shifting the sweep, is either by a hand lever, or by a rack secured to the sweep, and a pinion—on the starting wheel shaft—imparting the required effect; the former is used for small engines, and the latter for larger power. The means mostly universal for shifting the link—when the same is used—consist of a lever, worm, and wheel, or quadrant. In some instances, a screw and sliding block is preferred, but the attainment is the same in either example.

The crank shaft disengaging gear, in the event of a rupture of either of the paddle wheels, has met with much attention from those interested in the same; also, the expansion valve and gear. These two latter are adjuncts, however, and do not affect the arrangements, consequently, need no present attention, but will be fully digested in the chapter on details.

By the description herein given, it will be understood that the arrangements of the details of the oscillating engine, greatly affects the design or general arrangement of the whole.

The main points to be considered, at first, are the condenser and air pumps; after which the cylinders, entablature, lower frame, and crank shaft, should be noticed. In passing, it may be said, that the starting gear and valve motion must be duly remembered before deciding the principal distances of the main centres.

In examples, where the centres of the cylinders are imperative, the angles of the air pumps will be duly affected; but in either cases, accessibility to all the parts requiring inspection or repair must be the leading feature in the case.

The proper position of the snifting valve and cylinder relief valves should not be forgotten at the commencement. To be concise, all the details must be noticed before a design can be said to be correct, in imagination only, to say the least of it.

The illustration, Fig. 1, on page 33, represents a pair of oscillating engines with injection condensers. The cylinders are partially solid at the bottom end, with the necessary hole for the boring bar. This opening is securely stopped by a recessed door, thus forming a plain interior. The piston is hollow, of the metallic kind, fitted with studs and face rings. The piston rod is secured by a brass nut, recessed on the upper side, or top. The cylinder cover is cast hollow, with ribs to retain the requisite strength. The rod is guided in the stuffing box by a long bush and short gland of the ordinary kind. The slide valves are two to each cylinder, one on each side of the same. The kind of valve is that of the single-port type, packed at the back, with a metallic ring and gasket; the compression of the latter is attained by adjusting studs, and ratchet stops. The pressure of the steam is thus prevented from acting on the area of the valve, and thus mitigates the friction of the same; a communication from the back of the valve and the exhaust passage also assists to lessen the evil. The trunnion blocks and the lower

frame are in one casting. The brasses-cap for each block is adjusted by bolts and nuts; a cup on the top filled with oil ensures sufficient lubrication. The exhaust steam from the trunnions enters the condenser in the centre of the arrangement. The stuffing boxes are secured to the condenser, consequently are not affected by the motion of the cylinder. The condenser extends between the cylinders, below and beyond the same. The injection water enters, near the exhaust steam openings, to cause a sudden condensation. The air pumps—one on each side of the centre line—are angularly arranged. The barrels are fitted in suitable provisions: the top portion of each pump is separate, and supports the discharge vessel. The lower suction valves are situated on a perforated disc, secured at the extremity of the barrel, within the same. The pump piston is also a perforated disc; and on its top side are the valves, that act for the delivery. The final discharge valves and chambers are beyond the pumps, at the tops of the same—seen in the side elevation. The discharge pipe is on the top of each chamber, and projects within the same for a given length, thus forming a surrounding air vessel. The pump piston derives its motion from the crank on the intermediate shaft, and in the place of side guides, trunks are adopted. The connecting rods are secured side by side on the same crank pin, and prolonged to the lower ends of the trunks, where the connection is by a single pin and double eye projection, secured by a nut on the lower side of the piston. The feed and bilge pumps are secured on the front side of the lower frame: motion is imparted by the oscillation of the cylinder. The valve motion is that known as the back balance, single eccentric kind, with the loop to the lower end of the rod. A lever on the side of the discharge vessel, or hot-well, is connected by a rod to the loop, thus a ready disconnection is effected. The sliding sweep is manipulated by a rack and pinion, the latter on the hand-wheel shaft. The starting platform is beyond the hot-well. Separate gear is fitted to each engine. The expansion gear is the cam motion, with the shifting roller. The motion is rendered certain by a spring attached to a lever projecting opposite to that supporting the roller. The valve for cutting off the steam is the disc or throttle kind, hung in the casing secured beyond the trunnions—one of which only is seen in the illustration. The necessary supplementary valves, snifting, and cylinder-relief, are all properly arranged—the two former are shown in the side and end elevations. The mode of disengaging the cranks, is simply by the removal of the keys at the backs of the outer cranks. The entablature is in two portions, centrally connected. The section at the centre below the shaft bearing, is that of a parallelogram. The adjustment of the entablature is attained by keys, the top ends of which are screwed, and thereby shifted by nuts. The supporting columns are secured in the lower frame by keys: the top end of each column passes through the entablature, the final connection being made by a nut on the top of the same. The arrangement, now alluded to, may be considered as the ordinary mode of general practice.

INVERTED CYLINDER DIRECT ACTING SCREW PROPELLER ENGINES.

The firms in the Northern districts are famous for producing the arrangement similar to that illustrated by Fig. 2. The arrangement of the cylinders are much alike in all cases; but any variation in their design and positions is due to the particular type of condenser adopted, and in order to assist the definition of those circumstances, a brief notice of the adherent principles will be essential.

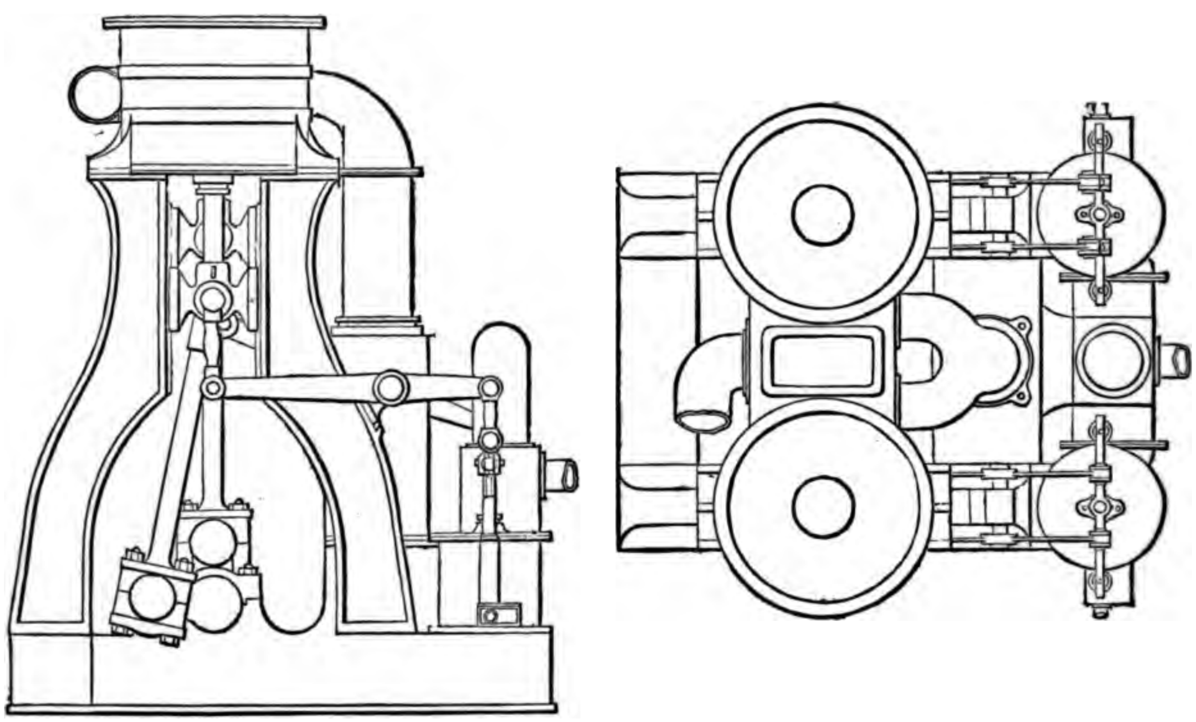
The vital points to be considered with the types of engines under notice, are the arrangement of the air pumps and valves, system of condensation, and the mode of working the pumps; and that the cylinders must be supported is imperative; therefore the next consideration is the applicability of those supports for other purposes.

When standards are used, the condenser is generally beyond the former; but should the condenser support the cylinders, a direct action for the pumps is often adopted. This latter is not, however, imperative, as the pumps can be secured at the side of the base framing.

Another matter also offers itself, worthy of comment: the position of the slide valves. Should their position be between the cylinders, the means for inspection and adjustment is sometimes cramped; while, if each cylinder is fitted with separate casings, valves, &c.—fore and aft—a free access for repair, &c., is ensured.

The modes of working the feed and bilge pumps must not either be forgotten; and it also becomes worthy of notice to consider their situation, for repair and inspection when requisite.

FIG. 2.



GENERAL INVERTED DIRECT ACTING MARINE SCREW ENGINES.

As before hinted, the condenser forms no mean portion of the engine; in fact, it is the principal cause for certain arrangements; because it is well known what are the restrictions imposed with condensation; they being that the air pump must perform its duty simultaneously with the entry of the steam; and suction and discharge valves are requisite in all cases, with access for inspection and repair.

The various positions for working the pumps are as follows. With a direct action, the pumps will be situated under the cylinders, with the centres of the former within the periphery of the latter. When the pumps are beyond—at the side of—the framing or condenser, a lever connection is requisite. The better mode is from the steam piston, but in either connection, the pumps have always a vertical action.

Thus far having briefly adverted to the principles adherent to the type of engine now alluded to, the following description of the most modern arrangements will complete the present section.

The illustration, Fig. 2—page 38—represents a pair of engines with injection condensers. The cylinders are supported on standards of a girder-like section. The slide valves are between the cylinders, but ample space is allowed for the requisitions attendant. The supply steam pipe is attached to the casing at the side, that for the exhaust being situated opposite. This latter pipe forms two separate connections at the top, and single at the end secured to the condenser. The expansion and contraction of the pipe in question is allowed, by the stuffing box seen on the condenser. The air pumps and valve chambers are secured beyond the condenser. Motion is imparted by levers connected at the one end, to links on each side of the connecting rod pin; the other extremity being in like manner secured to the crosshead; which is connected direct to the air, feed, and bilge pumps rods.

Side guides are sometimes used for the pump crosshead alluded to, but not imperative with short strokes and stiff gear. The injection valve is secured below the exhaust steam pipe, between the condenser and the air vessel, the latter being on the discharge chamber. The final discharge, water pipe, is secured to the outside of the chamber, and from thence to the ship's side. The guide-blocks for the engine piston rods are of the ordinary kind, with flat surfaces and adjusting pieces. The connecting rod is of the usual type and connection. The base or lower framing is of a box girder-like section, also forming a portion of the condenser and valve chambers at the side. The valve link motion is, of course, between the engines. The means for starting, stopping, and reversing, are attained by a hand wheel, worm, pinion, levers, &c. The supplementary valves—snifting, cylinder-relief, blow-through, and stop valve—are suitably situated. The doors, accessible to all the valves, are correctly situated without disarranging the surrounding portions. The cranks and shaft are in one forging, the bearings being fitted with adjustable brasses. The situations of this arrangement may be said to be the universal practice from the commencement to the present time. There are, however, a few,

with differently arranged condensers and mode of working the pumps, &c., to which allusion will now be made. When the condensers form the structure for supporting the cylinders, the former is at the extremities of the arrangement, or fore and aft. The air pumps—sometimes two to each engine—are secured to that side of the condenser, facing the centre line longitudinally. The valves are within the condenser and the doors for access are at the outer side. The feed and bilge pumps are secured to the lower framing. All the pumps are worked direct from the steam piston, thus dispensing with levers, &c. The slide valves are either situated at the extremities, or between the cylinders, the former position being mostly adopted. The guide blocks for the piston rods are the slipper kind. The guides are at the side most effective to resist the line of strain of the connecting rods, due to the direction of the crank's rotation. Other arrangements consist of the condenser being situated as the last, but the pumps worked by levers as before described. This latter mode is, perhaps, the preferable for access for repairs, but not for economy.

Surface condensers do not materially affect the position of the pumps in relation to those for the injection kind; the difference being chiefly an additional pump for circulation. The last arrangement is common to both injection and surface condensers: by continuing the latter on one side, for the entire length of the arrangement, in order to attain the tubular surface requisite.

In another example—the side frames are adopted, and the tube compartment, situated between the same, transversely of the hull of the vessel, or centrally of the engines. The requisite pumps are on each side of the condenser; motion being imparted to the former by levers. These latter are fixed at one end to a shaft or pin, the other extremity is attached to the crosshead of the steam piston rod, in the ordinary manner; and thus a reduced length of stroke is given to the pumps.

In some cases the pumps are of a combined arrangement; *i.e.*, one side of the piston circulates the water, while the opposite draws the condensed steam, at each stroke.

With a third classification, the condenser supports the cylinders centrally, the outer sides being sustained by wrought-iron stays. The requisite pumps are of vertical action placed below the condenser. Motion is attained by a long cross head secured to the steam piston rod—one or two being used—thus a direct connection is produced; with the advantage of a single guide being equivalent to rods, levers, links, &c. Another gain in this central position is, the correct distribution of the machinery in the hull, a matter worthy of due consideration. In order to reduce the longitudinal length of the arrangement—in proportion to the last example, the condensers are one to each engine, separately situated on each side of the same. Motion is imparted to the pumps as in the last example. The pumps are double acting, separately arranged in alternate positions on each side of the condensers. By this arrangement it can be readily understood that the access to all the requisite parts is effected with a certainty, without cramped room or discomfort. With re-

ference to the remaining portion of the different arrangements: each are much alike in principle of action, if not in design of detail, hence a repetition would be tedious, as well as superfluous.

The type of engine now alluded to has lately been introduced in our war vessels.

The Northern firms, as before stated, are prolific in their designs for this class of engine. The great cause for this monopoly of production is the confining data of the price attainable: primary commercial interests thus being the principal consideration. To the credit of the many firms engaged in producing inverted engines, a great advance has lately been made—both in design and economy of fuel, and with compound arrangements particularly.

HORIZONTAL ARRANGEMENTS.

DOUBLE TRUNK ENGINES.

The horizontal arrangement for screw engines has many originators, many friends, and of course, naturally, the usual amount of enemies. The object sought after, with the type in question, is compactness of arrangement, with free space above the cylinders and condensers. The length of the cylinders, mode of connection with the crank pin, and the form of condenser compose the longitudinal view. The arrangement in plan, and the area occupied is, regulated by the space between the centre of the cylinders and position of the slide valve. It may be added, in passing, that the outline of the condensers—fore and aft—rarely exceeds that of the cylinders. It can thus be understood that the beam of the vessel must have primary consideration before determining the longitudinal arrangement of the engines and condensers.

The connection of the steam piston with the crank pin is, a mechanical contrivance which engrosses the attention of all those who are interested in its effect. For land purposes, this matter is a simple supervision of freedom of access for adjustment and repair, which are generally at the command of the engineer; while, with marine examples, two essentialities ever present themselves: a given length from the cylinder to the crank pin, and the certainty of correct adjustment to all the requisite parts.

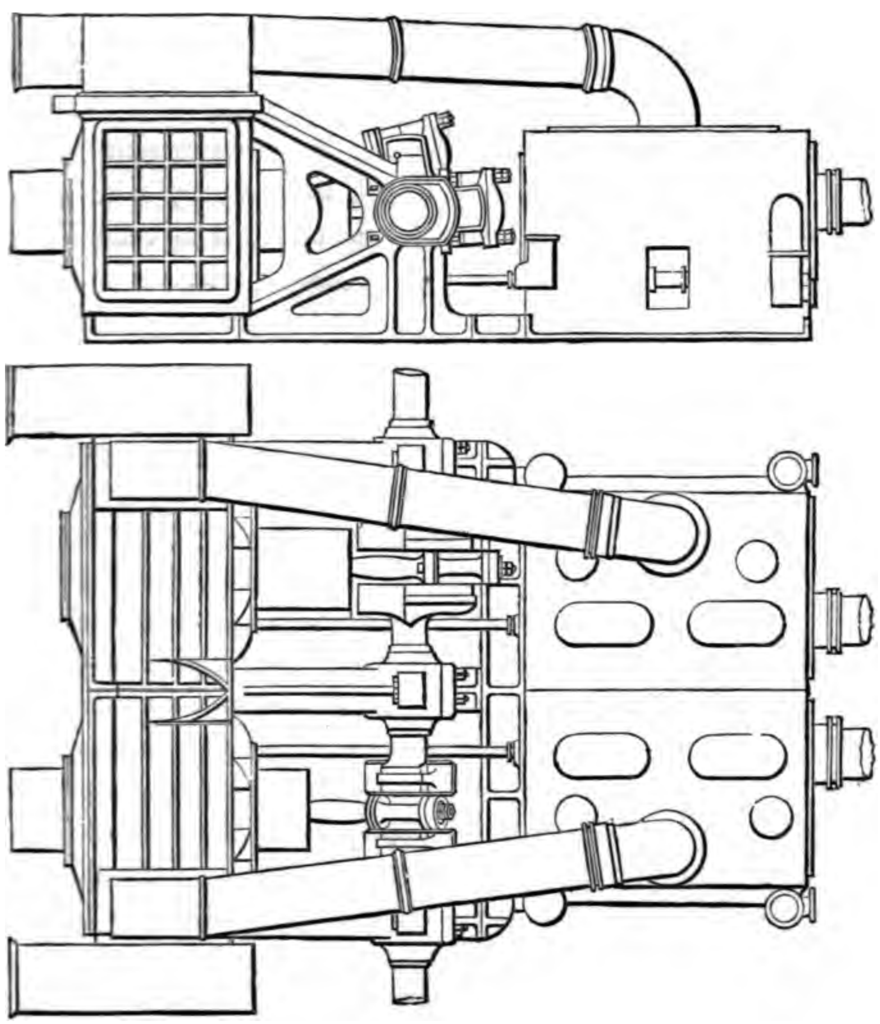
The advantages of the double trunk engine, also the simplicity of the transmission of the power exerted on the piston, to the crank shaft, next demand notice. The enemies to the horizontal arrangement held forth the argument that the cylinder, from liability to become oval, the effect must be conducive to the incessant piston-leakage of the steam. Whether this actually prompted the adoption of the double trunks matters but little, suffice it to say that the firm next alluded to, has succeeded in producing some first-class examples, combining simplicity of connection and access for repair, with superlative design and arrangement. The adoption of double trunks certainly is productive of equal guiding surfaces, a consideration, or rather an attainment, which marine engineers well understand. The absence of piston rods, outer guides, blocks, short length of connecting rod proportionate to the stroke of the piston, also, must not be forgotten by the friends of the trunk system.

Now having alluded to the advantages, and

quoted the fair view of the question, the theory of those opposed to the system under notice, must next be adverted to. The antagonistic question is this. Whether, the alternate exposure of the trunks to the atmosphere and steam, enlarged stuffing boxes, attendant friction, and the requisition of incessant lubrication of the trunks when in operation, if weighed against the advantages attained, do not turn the scale against them? To be concise, it may be said that "nothing is perfect," but simplicity of connection with mechanical contrivances, will always retain certain practical merits.

The illustrations—Fig. 3—represent an arrangement of engines by the firm of Messrs. J. Penn and Son. The detailed arrangements are as follows:—

Fig. 3.



MESSRS. PENN'S MODERN INJECTION CONDENSER DOUBLE TRUNK-CYLINDER SCREW ENGINES.

The cylinders are secured together on one side of the centre of the hull of the vessel.

The trunks are double, or one, on each side of the piston, passing direct through the front and back ends of the cylinders. The connection of the crosshead pin is attained by bolts and nuts passing through projections cast, with the piston and front trunk—the back trunk being a separate casting, and secured by studs and nuts. The connecting rods are of the ordinary single-end kind—adjustment being attained by the securing bolts and nuts. The main frames are of cast-iron, the caps of which are secured in a line with that of the cylinder. The cranks are counterbalanced by weights secured to the back of each, thus producing a uniform motion.

The slide valve latterly adopted by the firm now quoted is the equilibrium double-ported kind. The friction is greatly mitigated, by a recess in the back of the valve, which encloses a ring and packing, the outer side of the ring bearing against the cover of the casing, thus excludes the full area of the valve from being exposed to the action of the steam.

In other examples, adjustment is attained by set studs, ratchets, and springs: the two latter preventing looseness and leakage. The friction on the face of the valve is further obviated by causing a communication, from the circular recess at the back, with the condenser. The mode of imparting the motion to the valve is by the ordinary slotted link, and eccentrics. The valve rod—when one is used—is guided beyond the casing by a guide box secured to the main framing. When two rods are adopted, a crosshead connects the same. The guides are above and below the rods; the former being secured to the casing, thus dispensing with the guide connection with the main frame. The link adopted is that of the slotted type. The remainder of this motion is the ordinary kind, with brass bands and wrought iron rods.

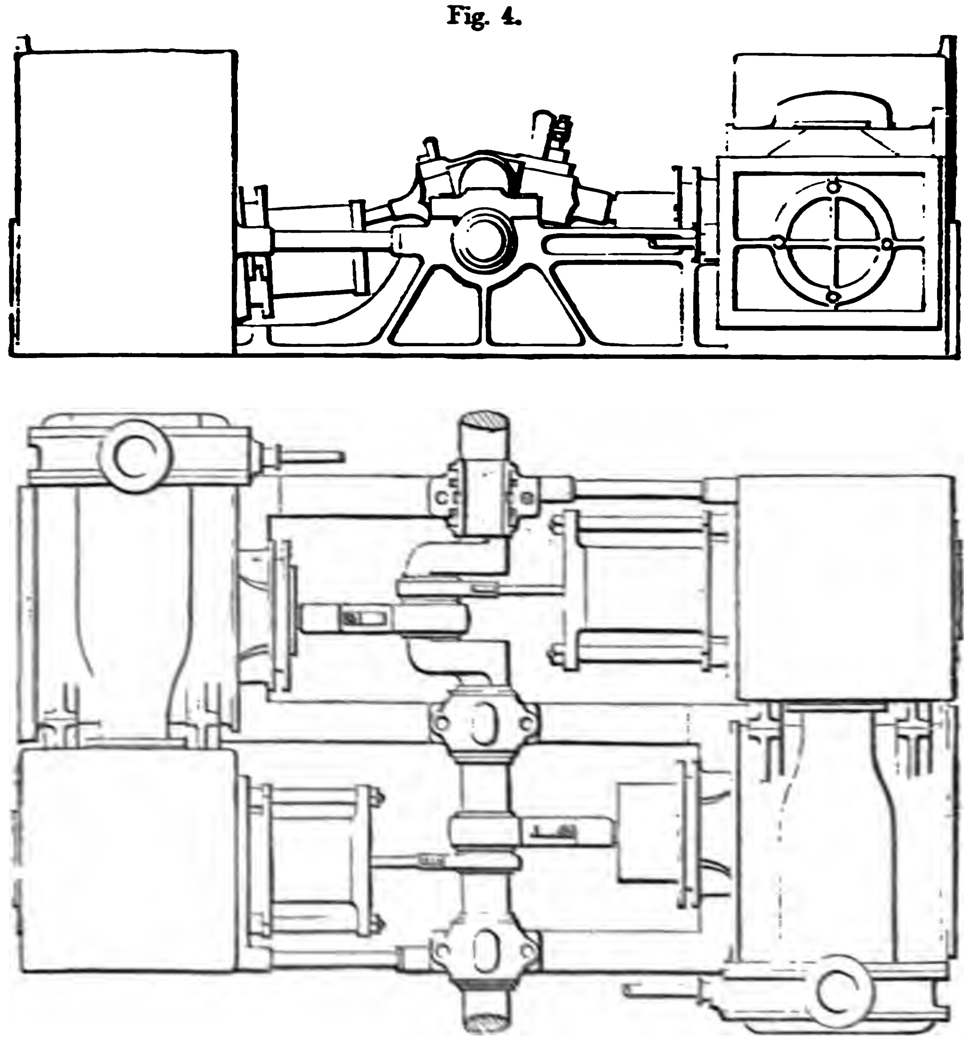
The mode of raising and lowering the link is generally by a rod connected to a lever, keyed on a shaft; on this shaft is secured a quadrant gearing with a worm, the latter forming a portion of, or keyed on, the starting wheel shaft. The connection of the lifting rod with the link is central, the firm seeming to prefer this to any other position.

The late mode of reversing, &c., has been thus: mitre gearing on the wheel shaft imparts motion to a rotative box encompassing a perpendicular screwed rod, the lower end of the latter being connected to a lever counterbalanced at the outer extremity. A connection with the link is attained by a rod attached to the lever in question, and thus any motion imparted to the screwed rod is transmitted to the link.

The supply steam is admitted to the slide valve casings through a separate bonnet or casing containing the expansion valve. The exhaust steam passes through separate pipes leading from the cylinder to the condensers. The latter, when of the injection kind, are generally arranged thus: the condensing and discharge chambers for each engine are in separate castings, centrally connected by bolts and nuts. The condensers are at the outer sides of the arrangement, or fore and aft of the hull; the discharge chambers being placed centrally. The suction and delivery valves are on the same level, above the periphery of the barrels of the pumps. By this arrangement, a more perfect vacuum is produced, than by the primi-

tive arrangement of the suction and discharge valves being under and over the pumps. The feed and bilge pump barrels are cast with the condensing chambers. The valves are attached in suitable boxes, secured to the exterior of the chambers forming the sides. Motion to all the pumps is imparted by rods directly connected from the steam pistons to the several respective pistons and plungers. The injection valves are secured at the side of each condenser, opposite or beyond each respective exhaust steam pipe. The cylinder relief blow through, and the snifting valves are all correctly distributed; and the connections to ensure ready manipulation are within reach of the starting wheel. The adoption of, and variation in, the arrangement of the surface condensers and expansive gear, by the firm alluded to, will be described in future under the proper titles heading the subjects.

Fig. 4.



MESSRS. RENNIE'S SINGLE TRUNK AIR PUMP AND TRUNK-CYLINDER SCREW ENGINES.

SINGLE TRUNK ENGINES.

To the firm of Messrs. J. & G. Rennie must be assigned the credit of the arrangement

shown by the illustration—Fig. 4—on page 44. In the place of the cylinders being connected on the same side of the keel of the vessel, it will be noticed that duplicate situations of each cylinder and condenser are preferred; being, in fact; a reverse position of the patterns, without the ordinary requisition of the usual alteration before moulding. The air pump pistons derive their motion from the cranks, also the feed and bilge pumps, the latter being connected to the former. The condensers are the injection system. The air pumps are the plunger type, single acting, without packing beyond the outer stuffing boxes. The suction valves are below the plungers and the discharge valves above the same. The condensing compartment is in the front portion of the casting. The discharge chamber is directly over the valves requisite to relieve the pump. The condensed steam passes—through a passage—over the top of the cylinder in each case. The slide valves and casings are situated at the side of each cylinder, fore and aft. The connecting rods are attached singly at each end. The adjustment of the brasses within the steam trunk is attained with free access, by an ingenious arrangement, thus:—the inner or smaller end of the rod in question is bored centrally from the slot to the length requisite beyond the outer end of the trunk. At this latter position a key or cotter is inserted. Adjustment is attained by a rod—loose within the bored cavity—bearing against the inner brass at the trunk end of the connecting rod, and the cotter beyond the trunk.

The mode of adjustment at the crank connection is, by the ordinary strap, cotters, gibs, and brasses. The air pump connecting rods are also similarly connected and adjusted. These latter rods are internally connected by two plummer blocks, secured to suitable provisions, situated about two-thirds of the length of the trunk or plunger from the closed end. The injection water enters each condenser at the side of the same. The doors necessary for access to the valves, are at the extremity of each discharge chamber. A provision is cast below each condenser to which the snifting valve is secured; it may be added that the condenser surrounds the plunger casing. The main frames can be readily understood from the illustration. The brasses are in four parts; the vertical adjustment is attained by the cap bolts. The lateral adjustment is produced by four keys to each bearing shown in plan, on the outer framing, the cap being removed to represent the means of adjustment. The feed and bilge pump barrels are cast with the condensers, on each side of the same. The valve boxes for the pumps are secured at the back end of each. The requisite motion for the slide valves is attained by the ordinary slotted link and double eccentrics. The mode of raising and lowering the link is by a pinion and quadrant, the former being keyed on the shaft of the starting wheel. The position of the starting gear is at the side of the condenser, the motion being transmitted to the links by levers and rods.

The cylinder relief, bilge injection, blow through, and other valves are all suitably placed, according to the requisition of each.

The position of the slide valve is not always the same, as shown in the illustration. In some cases the top of the cylinder is deemed

the most preferable place. The slide valves are usually of the double-ported equilibrium type. The packing at the back, is a ring recessed in a provision cast in the cover of the casing, instead of the body of the valve, as with those previously alluded to. Adjustment of the packing ring is attained by a gasket and set studs; and a communication from the space within the ring to the condenser, is effective of a further reduction of the friction of the faces in contact.

The type of engine now alluded to is also fitted with surface condensers. When those are adopted, the cylinders are connected together opposite the condensers, similar in plan to the cylinders of the double trunk arrangement. The expansion gear adopted by the Messrs. Rennie is simple and effective, and will be fully explained amongst the examples by the principal firms.

RETURN ACTING TRUNK ENGINES.

The transmission of the motion of the piston to the crank may be said to be the main effect to be attained. The previous examples of trunk engines alluded to are the direct connection type, *i.e.*, the cylinders and connecting rods are on the same side of the crank shaft: and the trunks are within the cylinder — the stuffing boxes, glands, studs, nuts, with the other requisitions being as ordinarily for piston rods. Now, with the trunks alternately exposed to the steam and atmosphere, many doubts have arisen as to the economical working of the types in question. Again, queries are in force as to the friction imposed by the enlarged glands and stuffing boxes: but all devices or disarrangements possess, more or less, certain evils, and it is but fair to say, that the evils discarded by the double and single trunk engines, exist in all other arrangements, viz., plurality of piston rods and guides.

The result to be attained, practically as well as commercially, is, the economical working of each type. Should the makers of each design or arrangement be capable of proving an inequality of the sums sterling of the working expenses, then the type that results with the lowest sum and highest duty must be considered the best production. On the other hand, expense of outlay must not be unnoticed. Capital, invested as a speculation, must not be compared financially with that attended with securities, therefore all those matters deserve lucid attention from the engineer as much as the commercial speculator.

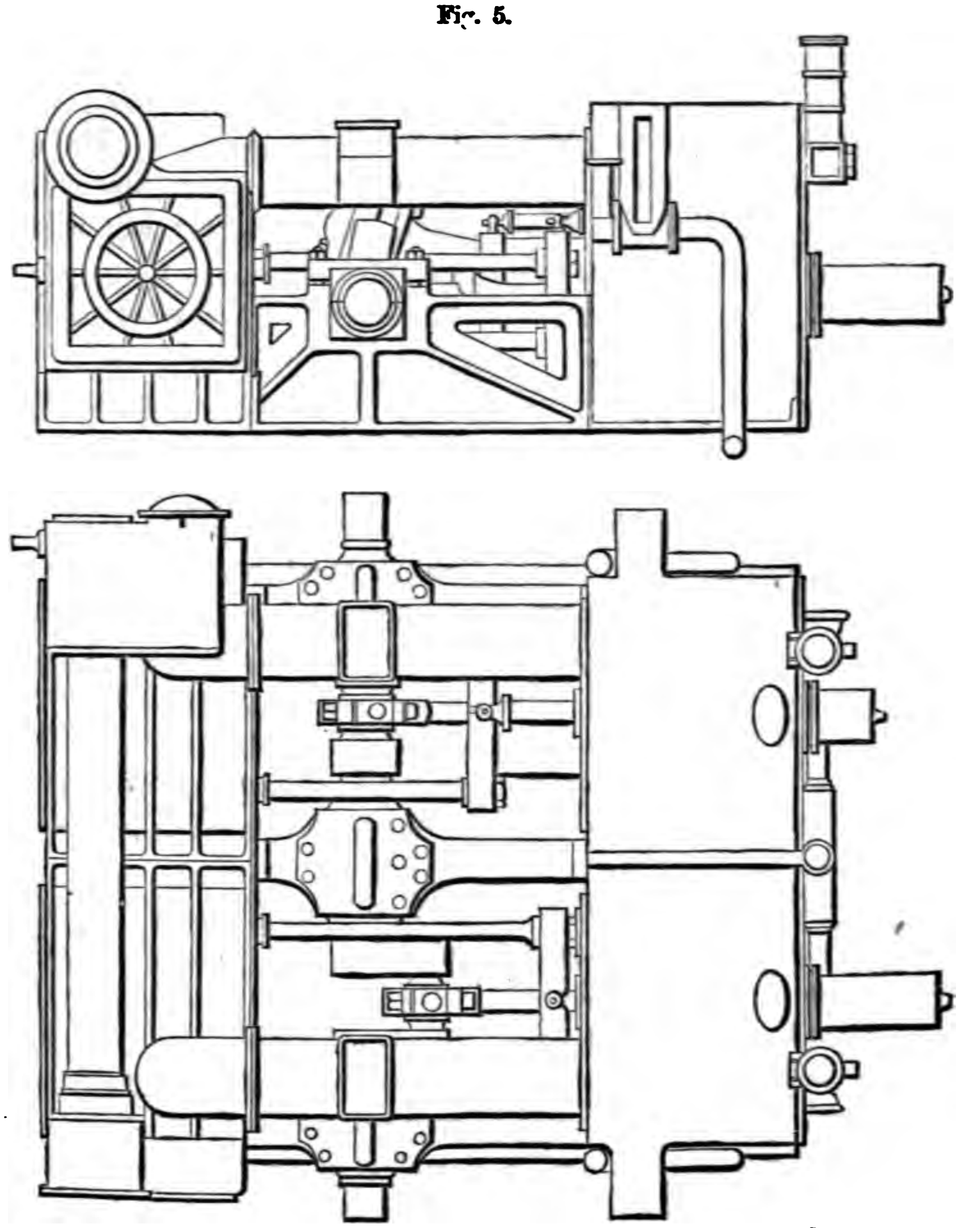
The arrangement depicted in the illustration, Fig. 5—page 47—represents one of the types of engines constructed by the firm of Messrs. R. Napier and Sons, Glasgow. It will be seen that—similar to the example last alluded to—the trunks are placed in the condenser. The mode of transmitting the motion to the cranks is, however, entirely different; in the place of single, or double, trunks in the steam cylinders, two piston rods to each cylinder are preferred, each passing over and under the crank shaft on either side of the cranks. The outer extremities of the rods are secured to the outer ends of the trunks, the latter forming the air pump plungers, also acting in the place of guides, &c.

By this it will be understood a perfect

eturn action is maintained with the merits nd demerits attendant.

The following description refers to the echanical arrangement of the component

Fig. 5.



MESSRS. NAPIER'S RETURN ACTING TRUNK AIR PUMP SCREW ENGINES.

etails. The cylinders are attached together n the same side of the crank shaft, and the ide valves are situated at the side of each ylinder, fore and aft; the valve adopted being the equilibrium double ported arrangement, packed at the back with the ordinary ring, spring, adjusting studs and nuts. The centre of the valve rod—unlike most examples by other makers—is above that of the crank shaft. The motion—for the valve—is imparted by the ordinary slotted link, eccentric rods, &c. The position of the link is at the

side of the condensers. The gear for starting, stopping, and reversing, is a screwed rod, with a sliding block encompassing the same; motion from the hand wheel shaft is transmitted by mitre gearing. The back portion of the sliding block—on the rod—is fitted into a guide, secured to the side of the condenser, to prevent lateral disarrangement. It may be added, in passing, that there are two hand wheels, one on each side of the condenser, keyed on the weigh shaft, which latter is supported on the condenser, passing across the top of the same.

By this particular position of the starting gear, a maximum length of eccentric rod is attained, while the cylinders are secured as near the crank shaft, as the length of the stroke of the piston will admit.

The main frames for supporting the crank shafting, are similar in principle and design to that depicted in Fig. 4—page 44. The caps are on the top of the framing, rather than in a line with the centre of action; adjusting keys are provided to prevent lateral looseness of the brasses.

The condensers are the injection kind, fitted with single acting pumps, the arrangement being as follows:—The steam enters the condensing chamber at the front end from the top; and the chamber extends midway above the top of the pump. Channels are formed on each side of the air pump barrel for the reception of the suction valves, those latter being inverted to ensure a perfect drainage of the condensed steam. The discharge valves are secured directly over those for the suction; and as the bottom of the discharge chamber is on a level with the top of the pump barrel, a perfect discharge of both *air* and water is effected. The chamber last alluded to is over the back portion of the pump, extending transversely for the entire width of the disposition. The injection water falls through a perforated plate.

The plunger, or trunk, is a cylinder, closed at the back end. The adjustment of the connecting rod is thus effected. At the back, or within the end of the trunk, is secured a single eye; the securing portion of this latter is fixed within a small cylinder, termed the adjusting tube, forming a portion of the trunk—projecting from the same equal to the length of the stroke of the engine. The securing end of the single eye is bored throughout its length, to admit a rod, the end of which acts against the inside brass. Adjustment is attained by the outer end of the rod being connected to the extremity of the adjusting tube. The connecting rod is forked at the trunk end. The connection with the crank pin is effected by securing bolts, brasses, and stop pins of the ordinary kind.

It will be noticed that the principle of the adjustment of the brasses within the trunk, with this example and that previously alluded to—Fig. 4—is alike; but the mechanical contrivance to attain the same result is entirely reversed. In the first example, the adjustment is at the front end of trunk; while in the latter case, the back end is deemed preferable. Each device, however, clearly illustrates the fact that thought and perception must have been in full force to produce the modes adopted, because the same result to be attained was always in view.

The supply steam enters the top valve casing secured on the slide valve casing of the engine nearest the boiler. The expansion valves are so situated, to be instantaneously effective. The expansion gear is the eccentric and semi-rotative valve: a complete description of this will be found in the chapter treating of the subject. The feed and bilge pumps derive their motion from the plunger or trunk of the air pump; and their requisite valves and boxes are at the back of the discharge chamber, fitted with the suitable springs and adjustable connections. The snifting valves are secured at the back of each air pump; underneath the latter is a passage from the condenser, by which a certain drain is attained. The doors, for access, to the valves within the chambers, are suitably arranged without disarrangement to the surrounding portions. The injection valves are at the side of each condenser, near the starting wheel, thus ensuring ready manipulation, without inconvenience—two attainments that can only be appreciated by those who are practically acquainted with the requirements. The cylinders are fitted with the necessary relief, and blow through valves; also lubricators are fixed at each end of the cylinder.

All the levers for manipulation are situated near the starting wheel; therefore, the arrangement reflects credit on the firm from which it emanated.

DIRECT ACTING
SINGLE PISTON ROD ENGINES.

The correct reference of the two words heading the title of this section, when relating to mechanical appliances, gives rise to divers opinions. Amongst engineers, generally, the words, "direct acting," is an acknowledged term for the connection of the piston rod with the crank pin,—by the connecting rod; while oscillating engines have received the title, and also trunk engines—each type being subject to the difference of opinion coincident with all popular matters.

Now, to define correctly, what is a direct acting engine—will be to first consider the arrangement of the relative details. With the oscillating engine, a *direct* connection, with the piston and crank is attained, but with the disadvantage of a vibratory motion together with a sliding action.

By attention to the law of forces, it will be readily understood that the requisition of two movements—in constant contact, to attain a rotative action—must be attended with an increase of friction; and, to still further analyze this fact, when the piston of an oscillating engine is at the bottom or top of the cylinder, the line of action—the line of connection must not be considered in the present case—is direct; but, immediately the vibration ensues, the disturbance of the straight line of action is caused; but, the connection will not, in any form of movement, be altered; hence, when an oscillating engine is said to be "direct acting," the term "direct connection" will be the more appropriate—or, rather, the more explanatory.

The single and double trunk systems now claim attention as to the correct title, due to the action of their connections; of course, those engines are directly connected, but as "direct acting" types there certainly is a doubt, thus causing the requisition of an

analytical investigation. Now it may be argued, that with the types under notice, the transmission of the power is direct, due, of course, to the connection of the piston with the crank pin. It next becomes necessary to consider whether the piston's action, with the trunk, is as truthfully conveyed with the connecting rod, as by the ordinary piston rod. It is very certain that the concentration of the steam force on the piston is an advantage, and therefore, with the ordinary single piston rod engine, the power is the most directly transmitted; while, with the trunk engine, the force exerted by the steam against the piston is by no means concentrated at once, but rather expended at right angles from the face of the piston to the crosshead; and, however difficult this may seem to the young engineer to understand, it can all be made clear by thought and a practical acquaintance with each type in question.

The "direct acting engine," actual, is a cylinder whose piston has a rod centrally secured, of the requisite length proportionate to the stroke—the stuffing box, guide block, and guide, being situated between the front of the cylinder and connecting rod, and the attachment of this rod to the crosshead, are matters of the highest importance; but, as this is not, perhaps, universally considered amongst young engineers, a little explanation of the principles will not be out of place.

The crossheads of all types of engines are for one purpose virtually, but actually often made to perform a plurality of attainments. Now the line of strain, or, rather power, is in the central portion of the piston rod; it is then obvious that the connection should also be central. A perfect connection can be attained with the ball and socket; but as, however, that means of adjustment is far from perfect, it is rarely, if ever, used for marine engines. A long crosshead with side guides is not correct in principle; although, when of sufficient strength to warrant non-deflection, it is admissible. The connection of the crosshead with the piston rod is often by nuts, in other cases with a cap and double attachments; the latter mode is adopted to attain a central connection of the connecting rod, to which latter rod the entire force is concentrated.

To still further define the principles now under notice, will be to advert to the strains imposed by the angles of the connecting rod. The crosshead, be it remembered, has a sliding motion, directly communicated from the piston rod. The connecting rod—at the crosshead end is similarly exerted—is connected to a crank pin, the motion of which is circular. The line of strains will, at all angles, be centrally of the line of connection, hence it is obvious that the greater the angle the more the friction imposed.

The lateral strain is almost neutral in proportion to that produced by the angle of connection; but, nevertheless, the actual effect exerted should not be unnoticed; and to counteract the effect of that strain, sides or flanges to the guide blocks are introduced, while the frictional wear is little or nothing with stiffly proportioned details.

Referring next to the direction of the motion of the crank pin, which is of great consideration in marine engines, and although apparently simple as the cause and effect may

be, a brief allusion to the fact, that the friction on the guide block can only be on one face at the same time, must not be deemed superfluous or even tedious by students,—which, by the way, we all are more or less.

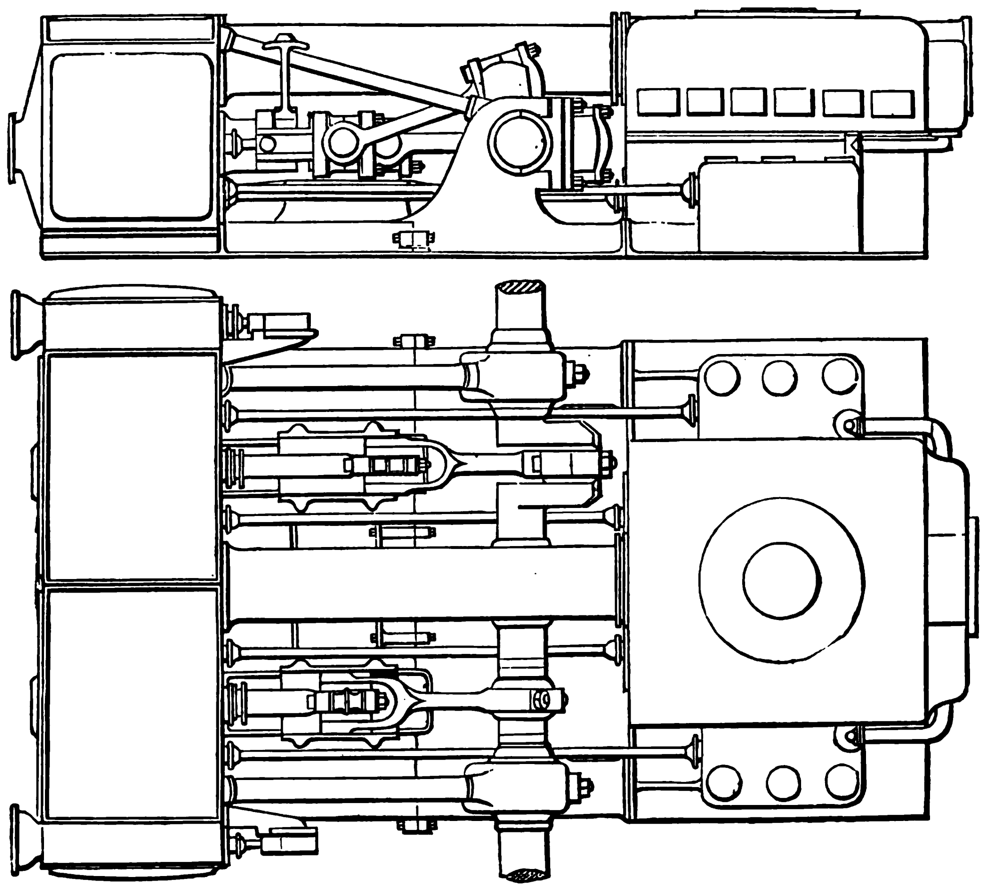
To clearly understand the strains and the effect of the angles of the same, will be first to understand the effect of the action of the steam against the piston—presume the piston to be at the back end of the cylinder, the crank will be, on the horizontal line with the pin, nearest the cylinder. Now, the next point for consideration is the movement of the crank—remembering that the pin is a dead point unless under the effect of momentum, when actually acceleration is attained by centrifugal force. Again, assume the crank pin to be ascending, the piston, be it remembered, is actuated direct by the pressure of the steam. Now, if it is assumed that the force of the steam be concentrated into one line, it can be readily understood that—the crank pin being a dead centre—the angle assumed by the connecting rod, and the strain therefrom imposed will be on the lower face of the guide block. To still further test this proposition, conclude that the crank pin is at half stroke, or rather, the piston is at that point of progression; it is obvious that the connecting rod will be at the greatest angle. The piston rod is now being impelled to the utmost point, and although the acuteness of the angle of the connecting rod is increasing in proportion, the principle of the strains imposed is unaffected. The crank is now presumed to be at the reverse position, but horizontally situated, as at the commencement. The crank pin is descending, the strain on the piston rod is tensile, but the effect on the guide surface is the same throughout. To reverse these strains, and the position of friction, the crank must be forced in a contrary direction at the commencement—as before stated, the crank pin must be regarded as a dead centre, or point; it is now added, in passing, that the connection of the piston and crank pin being virtually *direct*, the strains imposed are due only to the departure from the straight line of connection.

The truthful evidence of the laws of the forces, now under notice, can be proved by a simple mechanical contrivance. Let a rod, horizontally secured within guides at the one end, be connected at the other to a similar rod secured at the outer extremity at an angle to a fixed joint. Now, on pushing the first rod, the connection—of the two rods—will leave the plane line below the same. On pulling the rod in question, the jointed portion will ascend from the centre of movement, and thus clearly pourtray the strains imparted on the upper and lower surfaces of the guide block. The fixed joint represents the centre of the crank pin at half stroke: it can therefore be readily understood that the analogy is perfect; the crank pin being actually a dead or fixed point until the inertia is overcome. The shifting connection represents the guide block and cross head, therefore the actual tendency, or, rather, cause of the ascent or descent of the same from the plane line is conceivable. The guided extremity of the first rod represents the piston.

Having thus far treated of the principles of the "Direct-acting Engine," attention will now be given to the arrangement of the same, selecting an example by engineers of well-known repute.

The firm of Messrs. Humphrys and Tennant have long since been famous for producing the form and disposition of detail depicted by Fig. 6, below. The arrangement can be said to have been carefully deduced; presenting, also, a plain exterior, both pleasing to the eye and practical to the mind. The following description will assist to further appreciate the many points worthy of notice.

Fig. 6.



MESSRS. HUMPHRYS AND TENNANT'S DIRECT-ACTING MODERN SCREW ENGINES.

The cylinders are secured together on one side of the crank shaft. The slide valves are the well known double ported equilibrium type, packed at the back with a ring and gasket. The pistons are of the metallic kind, cast hollow, with ribs to retain the requisite strength, while under pressure. The piston rod is secured by a nut at the back of the piston, centrally of the diameter. The stuffing box is the ordinary kind; the gland being adjusted by studs and nuts. The lubrication of the piston rod is maintained by an oil cup, or channel, encompassing the rod beyond the packing gland, forming a part of the bush in the latter. In order to prevent the oil wasting by the motion of the piston rod, a stuffing box and gland is beyond the oil chamber, and thus economy of lubrication is ensured. The form of the rod under notice, at the connection with the guide block, is as the letter ⊣ on its side.

The firm in question can lay claim to special notice for the guide block they have introduced, which is in accordance with the direction of the movement of the crank pin, and the strains resulting therefrom.

The block, now alluded to, is in halves, each portion being almost a duplicate of the other. At the underside of the block—below the piston rod—flanges are cast; underneath these latter, is connected a plain surfaced addition, termed the slipper, or, as some authorities have it, the shoe. The line of contact of the slipper and the block, is at an angle; thus adjustment can be ensured: a stud and nut being introduced for that purpose. The section of the guide channel is of a double bracket-like shape—the under side being connected. The connection of the connecting rod is attained by the latter being of a forked form clasping the sides of the block; the pin, forming the attachment, is centrally situated in the block. The adjustment requisite is attained by securing bolts, over and under the block pin; the strains imposed on these bolts are, of course, equivalent to that exerted on and against the piston rod, therefore the areas are equal. The nuts of the securing bolts are prevented from looseness of contact by stop rings, pins, and outside keys. The lubrication of the block pin is attained by a wiper and can, the latter being situated at the requisite height above the former. The crank end of the connecting rod is semi-solid, adjusted by securing bolts, similar to those for the guide block. The brasses are circular, retained, by their contact with the securing bolts, and thus the angular seats are obviated. Suitable flanges prevent lateral movement, and lubrication is obtained by the wiper and suspended can as for the guide pin. The cranks and shaft are in one forging of plain exterior, consistent with uniformity and strength.

The main frames next claim attention. In the present case, those details are fair examples of strength, and simplicity of construction; being of cast iron, forming, at the base—between the cylinders and crank shaft—a floor, that is connected to the condenser centrally of each bearing.

The guide channel—for the guide block—is secured by studs and nuts, being a separate casting. Between the guide channel projection and the supports for the crank shaft, a transverse connection of the framing throughout is introduced, in order to ensure a perfect casting and ready transmission of detail. The brasses in the main supports are the ordinary kind, adjusted with securing bolts and nuts. It will be noticed that the frames, between the cylinders and the crank shaft supports, are raised—proportionately to the preceding examples—but very slightly from the base line, at the connections with the cylinders and the condensers. In order to resist the direct strain—above the centre of the crank shaft—a stay is secured between each frame to support the front of the cylinders, and by that connection, the requisite resistance against the side strains are assisted. Each of the main bearings is provided with oil chambers and water tubes—the latter being required only in the case of heated bearings.

The correct relative positions of the condenser and air pump form a considerable portion of the difficulties ever before the marine engineer; because the attainment of a

good vacuum is always desirable, and indeed imperative; therefore, the position of the condenser demands much attention.

It will be remembered, that the condenser represented by Fig. 3—page 42—is over the air pumps, but each pump drains separate compartments. Now, with that arrangement two exhaust steam pipes are requisite; while duplicate connections must be provided, but with the advantage of separate arrangements, which, in the case of a rupture or defect in either portion, is of considerable utility.

The arrangement of the air pump and condenser, by the firm now alluded to, is worthy of notice, not only for compactness of detail, but also for the correct position of the pumps and condensing chamber; and in order to render the following description of practical as well as of theoretical value, an imagination of the sections must be considered; suppose a transverse section first. The air pumps, double acting, are near the base line; of such a distance apart as the periphery of the steam cylinders determines. The condensing chamber—in one compartment—is directly above and between the centres of the pumps; in fact, the sides of the chamber and the centres of pumps may be taken as the same, vertically. The exhaust steam enters the chamber, through one pipe below the roof, at the front end. In order to ensure a discharge of the condensed steam—through the exhaust pipe—a suitable projection is secured at the opening in the condenser. The injection water enters the condenser at the back end, the correct distribution being attained by a perforated pipe suitably secured in the chamber. The suction valves are secured in the bottom of the chamber, inverted in action, to ensure thereby a correct drainage—which attainment should never be overlooked. At the outer sides of the pumps,—or fore and aft,—above the same, are the discharge valves. These latter are situated in the discharge chambers, the floors of which are above that of the condensers. The chamber now in question forms a compartment on each side of the condenser, being connected at the back end of the latter, to attain the final discharge. The suction and delivery valves being above the pumps, it can be understood that the transverse width of the entire arrangement can be contracted, while the longitudinal position of the valves throughout, makes good the requisite area. The doors for access to all the valves are at the sides of the discharge chamber, and the top of the condenser.

The feed and bilge pumps are at the sides of the air pumps, in separate castings. The plungers are of the piston kind, thus producing double action. The suction and delivery valves are over and under the pump barrels, but in principle of action the same as those for the air pumps. Doors on the top and ends of each chamber are situated for internal access to the internal requisitions.

The valve link-motion for marine engines has engaged the attention of those who understand what is required from the same; and at the present stage of improvement many arrangements are used, but the *actual* attainment in each case is not what it should be; because what is requisite, is a correct action, combining strength with lightness of material in weight; while it is, perhaps, a difficult

task to produce an example of mechanical contrivances, showing more evidence of thought, than the arrangement of the valve link-motion of the marine engine. Our marine engineers have very clearly carried out their ideas on the subject; and, in fact, have actually put in practice the results of their imagination congenial with science; which, although creditable in most cases, it is not too critical to state that, if much has been done, more is required.

The class of link-motion by Messrs. Humphrys & Co. is really an almost isolated production, but wanting perfection, as with all mechanical attainments. The link is solid, connected at the extremities with the eccentric rods. The block—encompassing the link—vibrates in suitable provisions, adjustment being attained by studs and nuts. The levers and connections for raising the link are below the centre of the plane line of action, and a top guide for the upper extremity of the link is introduced; being, indeed, almost imperative. This guide is supported on a hollow, box-like provision, secured in front of the valve casing and cylinder; which is also a guide channel for the valve rod. It will thus be understood that the raised position of the link is preferable to attain the more correct action, due to the arrangement of the details.

Controversies innumerable may be said to have been brought forward as to the "correct situation of the starting wheel." Some marine engineers state that the top of the condenser is the most convenient position; others, equally wise in this question, prefer the side of the arrangement. Now, to decide hastily in either case, will be unfaithful to the cause, therefore a brief synopsis of the requirements will not be out of place.

It is, of course, well known that, with vessels of war, sudden and certain manipulation is of the greatest importance; hence the numerous examples introduced to attain the same effect.

The power requisite to move the slide valves of some of our large marine engines is considerable; and, granting that the effect of the steam on the back of the valve is greatly mitigated by the various contrivances, yet the friction must be great. The primary consideration of the designer of the starting gear for marine engines should be "time and power," *i. e.*, how many revolutions of the hand wheel with a given force will raise or lower the link, and thus alter the position of the valve. The position of the starting platform will, also, to a great extent, regulate the type of gear employed.

Now as to the correct locality. Much depends on the arrangement of the engines and the condensers; it is, therefore, impossible to lay down a practical rule, without some exceptions, in this case. Some makers prefer the top of the condenser as the best position for starting, insisting that the engineer in charge can better perform his duties at that place than at any other; other authorities make use of the top of the cylinder for the support of the platform. In other cases, the side of the condenser has been deemed the most safe and convenient position. By this description of the result of many opinions on the same subject, it can be readily understood that each maker or designer of marine engines maintains his ideas throughout the entire arrangement of the details.

With the arrangement of the starting gear by Messrs. Humphrys' great originality has been carried out, and it is very evident that the gear in question had primary consideration with the remaining important portions, the connection of which will be appreciated by the following description.

Presuming a side elevation of this gear —centrally of the length of the feed pump chamber is secured a shaft, internally fixed by a nut; a tube, forming also a pinion, is supported on the shaft. The starting wheel is secured on the tube, and thus the motion imparted is non-effective to the shaft in question. This pinion gears with a toothed wheel situated below, supported by a second shaft, and alike secured at the first. The toothed wheel has formed with it a second pinion, which latter gears with a rack, horizontal in action. This latter detail forms the extremity of the connection with the lever on the main weigh shaft, on which latter are the lifting levers connected to the links. The mode of action, therefore, will be thus:—On motion being given to the hand wheel, the pinion likewise affects the lower wheel, which latter, by the second pinion, shifts the rack, and thus the link is raised and lowered. The principal gain by this position for the starting wheel, in the present case, is the simplicity of the gear introduced: there being an absence of brackets, screws, mitre gearing and vertical shafts.

Those latter portions of mechanism are not here condemned as entirely objectionable, but rather that a plurality of details should be, if possible, avoided in marine engines.

The remainder of the component parts, or, rather, the supplementary details of the arrangement under notice, do not present much to prompt further description; suffice it to say, due attention has been given to the vital consideration, *i.e.*, the attainment of perfect manipulation of all the requisite valves without shifting from the direct situation of the starting wheel.

RETURN CONNECTING ROD.

DOUBLE PISTON RODS ENGINES.

THE types of engines previously alluded to,—with the exception of Fig. 5—page 47—are arranged, with the cylinder and connecting rod, on the same side of the crank shaft. It has been proved in the previous section in this chapter, that the arrangement of details therein alluded to, is the most correct in principle. It now becomes necessary to observe the proportions available for given beams of vessels.

With Double Trunk Engines, the greatest length from the centre of the crank shaft—transversely of the hull—may be taken as follows:—Throw of crank, + length of connecting rod, + length of back trunk.

For Single Trunks the proportion will be deduced by: throw of crank, + length of connecting rod, + distance from connecting pin to back of piston, + clearance and width of back end cover.

In the case of Direct-Acting Single Piston Rod Engines, the proportion will be produced thus:—Throw of crank, + length of connecting rod, + distance from centre of block pin, to heads of securing bolts, + clearance and total depth of glands and stuffing

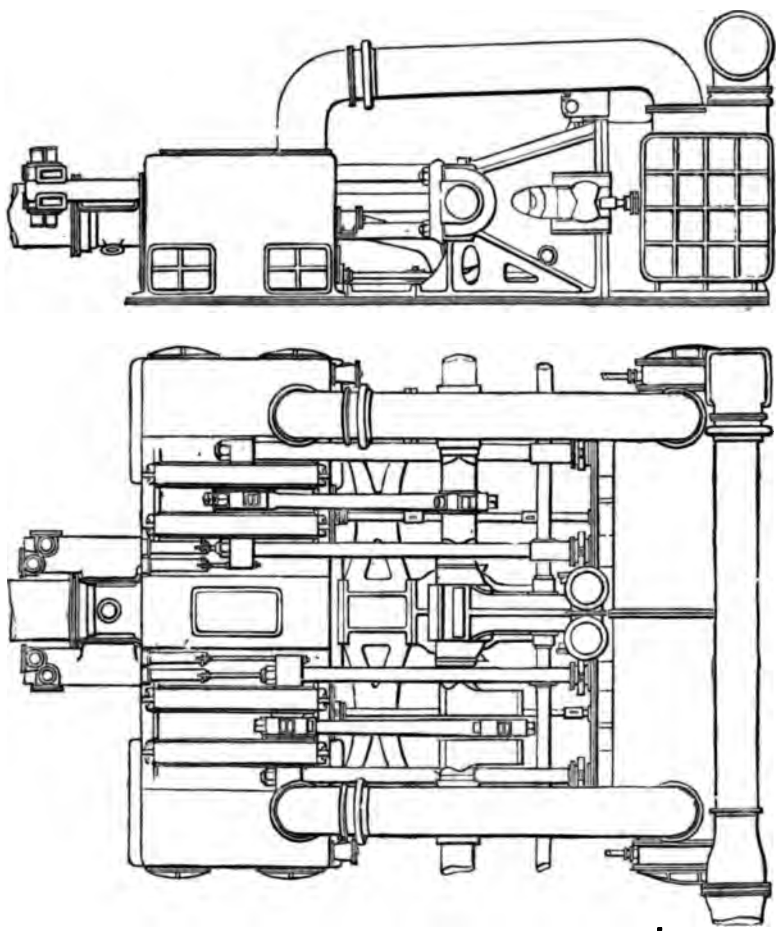
boxes, + length of stroke, + depth of piston and clearance on each side + depth of back end cover.

To deduce the length of the arrangement next under notice—Return Connecting Rod Double Piston Rods Engines:—Throw of crank, + length of connecting rod, + half length of guide block, + remaining portion of the guide channel, will determine the limits.

Now from those calculations — simple addition—a correct proportion of the length of the space occupied—transversely of the hull—can be readily understood. It must be added that—free-hand scaleless—diagrams of any type *cannot* truthfully pourtray the actual proportion; and that an observation of dimensions will in all cases produce correct results; which, perhaps, not always gratifying, are nevertheless reliant.

From the proportions now alluded to, it will be understood that the ratio of the connecting rod's length to the throw of the crank is the main consideration, and obviously the cause for the adoption of the type.

Fig. 7.



MESSRS. RAVENHILL'S MODERN RETURN CONNECTING ROD SCREW ENGINES.

The advocates of the direct-acting engine will, of course, lay claim for especial attention to the correct connection and compactness of that arrangement; while further maintaining

their argument by the fact that a maximum guide block surface, shortness of the piston rod, and non-interference with the space allotted for the condensers, are matters sufficient to render the arrangement in question worthy of general adoption.

The force of argument quoted by the promoters of the return action type is simply the attainment of less longitudinal space occupied, with equivalent power exerted, to that by any other arrangement. It will thus be understood that the gain in this case is condensed into one great fact, rather than a plurality of minor acquisitions.

Now, to describe the illustration, Fig. 7—page 57—this arrangement represents a pair of return connecting rod double piston rods engines, by the firm of Messrs. Ravenhill, Hodgson, and Co.

The cylinders are secured together on the same side of the keel of the hull. The main frames are the design general with that firm. The flange connected to the cylinder is prolonged from the base to the top of the latter, thus dispensing with the stay seen in some of the previous examples. The main caps are secured by bolts and nuts in a line with the strain exerted by the piston rods, thus requiring direct end adjustment only. The cranks and shaft are forged together, the shoulders of the bearings being represented, or formed by loose collars, secured at the inner ends of the outer brasses. Each main frame is sustained in a direct line parallel with the centre of motion by a distance piece—in one casting—connected to the foot of the frames and the projections on the condensers. The connecting rods are the single end type, fitted with adjustable brasses and securing bolts and nuts.

The connection and guidance of the piston rods are attained by a wrought iron crosshead, centrally turned, which is bent vertically in opposite directions at the extremities, to receive the piston rods; these latter pass over and under the crank shaft, on each side of the cranks. The connecting rod clasps the turned portion alluded to. The guide blocks, one on each side of the connecting rod, are secured to the crosshead, and work in open channels; these latter being adjusted by bolts and nuts at the extremities. By this arrangement of the guiding portions, a firm motion is preserved.

To follow the details consecutively as arranged:—the feed and bilge pumps are secured on each side to the final discharge water pipe. Motion is imparted to the pumps by the plungers being connected to the crossheads, in a line with the inner piston rods.

The condensers and air pumps next require attention. Presuming an end view of the arrangement, the description will be thus: the condensers are raised from the base at the extremities—fore and aft—or outside the guides for the piston rods. The exhaust steam enters the top of the condensing chamber, at the front end. The injection water flows through the valve box secured at the back end of the chamber; internal dispersion being by a tubular spray pipe. The snifting valve, secured at the back end, is near the bottom of the chamber, which latter is level with the lower line of the final discharge pipe. The suction valves are inverted, for correct drainage, and assist the action of the air pumps.

These latter are double acting, situated at the base line—the peripheries being below it—directly under the inside guides for the piston rods: motion is imparted direct from the steam piston. The discharge valves are on a level with those for the suction; and the final discharge pipe, before alluded to, is common to both arrangements, being centrally placed.

Now—to return to the primary details that produce the effect of the whole—the slide valves are of the equilibrium type, double ported, having narrow openings for the supply and enlarged ports arranged in the cylinder facing for the exhaust. The position of the casing, it will be noticed, is at the outer extremity of the cylinder, rather than centrally placed. This uneven position is preferred, to retain a given length for the eccentric rods. The link-motion adopted in the present example is a solid link, with overhanging or outside connections. The eccentrics transmit their motion in a direct line with the valve rod. The link at the inside, next to the main frame, is supported in a block, the latter vibrating in a suitable portion that is guided at its back. The means adopted for raising and lowering the link is by a screwed rod, and a slotted lever clasping a block, the latter deriving its motion from the screwed rod. The primary motion is received from the hand wheel situated on the top of each condenser. Mitre gearing—on each side of each exhaust steam pipe, at the front of the cylinders—imparts the required rotation to the screwed rod. It may be added that each engine has separate starting gear.

Remarks, to the present, it will have been noticed, relating to starting gear, refer especially to manipulation, or rather the adoption and reliance on the effect of the same. Now, as before stated, "time and power" should always be the consideration first in question with the effect of mechanical appliances. Marine engineers being, of course, cognizant of this natural—but simple—fact, have duly considered the matter, hence the production of *separate engines* for starting, stopping, and reversing those used for propulsion.

The firm now alluded to have introduced steam cylinders—vertically arranged—central of the main cylinders, at the front. These minor engines lift the links, by their connection with respective levers on the weigh shaft. A perfect control of the steam, entering these cylinders, is attained by the requisite valves and rods, the latter extending to the starting platform on the condensers.

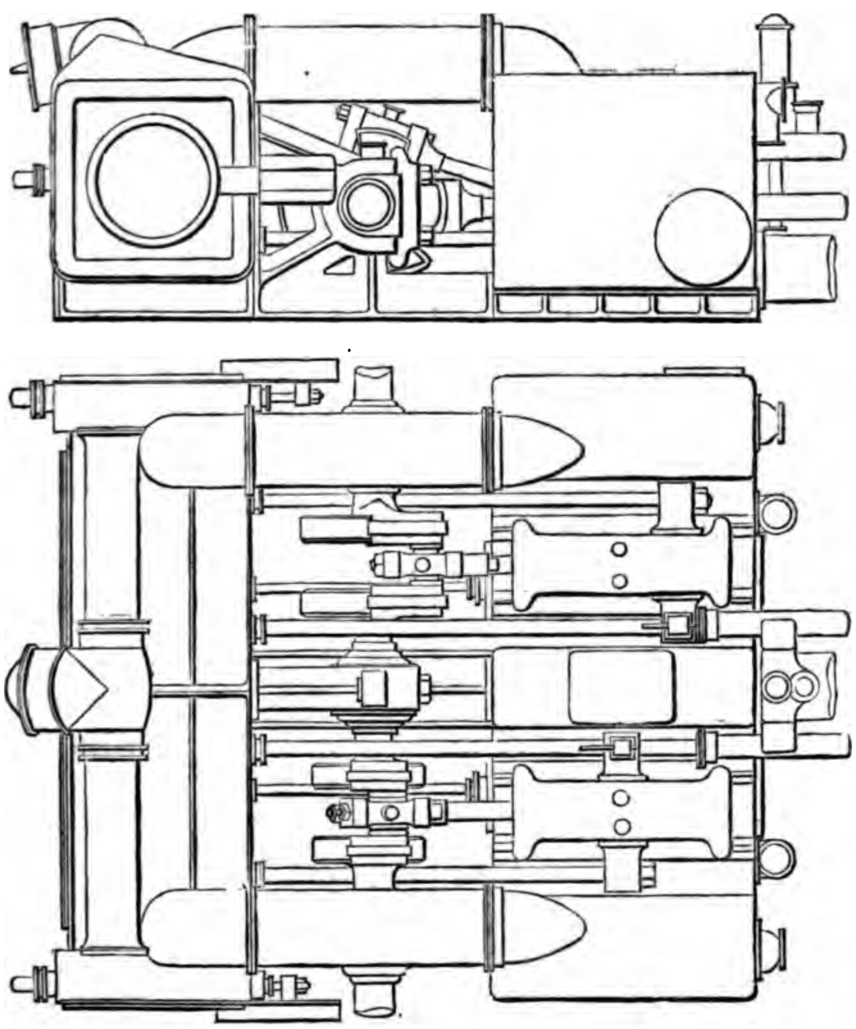
The main supply steam enters the main valve casings, through a pipe secured over, at the back part of the cylinders. The requisite blow through, relief, and stop valves are all respectively situated in their correct positions, with ready manipulation attainable at the desired locality. The expansion valve and gear adopted by the firm under notice will be duly treated and described under the heading—"expansion gear."

SINGLE GUIDE RETURN CONNECTING ROD ENGINES.

The effect of the angularity of the connecting rod on the guide surface, above and below the piston rod, is not perhaps universally understood. The strains imposed will, in all examples of arrangement, emanate primarily from the piston, and the force exerted be due

to the area of the latter being operated on by the pressure of the steam. It is, of course, patent to all, who understand mechanics, that the centre line of transmission of power is the correct line of resistance. For example: the steam by its action on the piston imparts a direct motion to the crosshead of the connecting rod. The power may be said to be communicated throughout the piston rod's length alike, but on the force reaching the connecting pin, a point of rest interposes, and thus checks the transit of the force. Now the line of force, as before stated, is central, and it has been previously stated that the lateral friction is almost neutral proportionately. On due recognition, then, of these facts, it is obvious that the line of resistance will be more effective when centrally located, than when on each side of the line of primary action.

Fig. 8.



MESSRS. NAPIER'S MODERN RETURN CONNECTING ROD SCREW ENGINES.

The advantage of a central guidance for the crosshead of the piston rods is duly recognized by the firm of Messrs. R. Napier and Sons; Fig. 8, above, being

a plan and side elevation of that particular arrangement.

The arrangement of the condensers and air pumps is similar to the last example described. In the present case, much less space transversely between the condensing chamber and that for the discharge can be maintained by the adoption of the central guide. The condensers are beyond the piston rods—fore and aft—the suction valves being inverted, in the floor of the chamber. The air pumps are placed centrally of the arrangement—the distance between the centres of the barrels being accordant with the peripheries of the steam cylinders. The discharge valves are arranged to admit the final discharge pipe being near the base line, and thereby reducing the height of the air chamber; at the sides of which are secured the feed and bilge pumps, whose motion is produced by a direct connection with the inside piston rods, which latter pass over the crank shaft. The valve boxes, air tube, and relief valve are situated at the back of the final discharge chamber, on each side of the main discharge pipe.

The guide channel for the piston rods crosshead is a planed portion—for the lower surface—secured on projections at the required position. The top surface is separate, secured at the extremities to vertical standards by bolts and nuts. The guide-block may be considered as a box, open at each end; the hollow portion being arranged to receive the connecting rod and the crosshead-pin. The adjusting provisions are suitably arranged, with the recognition of the effect due to the direction of the crank pin's movement.

By following descriptively the consecutive portions of the arrangement now under notice, the cranks and shaft will next be alluded to. The details in question are in one forging, with counterbalances at the backs of the cranks, to retain a uniform motion. The main frames present nothing worthy of much comment, beyond that given to the previous examples, the principle of connection and adjustment being alike in each respective case.

The steam cylinders, in the present example, are secured similar to the last, with the exception that the proportionate space between the centres is a lesser dimension. The exhaust steam passes direct, separately, from each cylinder to the condenser opposite. The slide valve and casing are of the ordinary type, before alluded to. The injection valves are secured to the back of each condenser. Those for the bilge injection are secured at the side of the condensers. The snifting valves are secured at the back of the connection chambers—or passages—near each air pump. The blow-through valves are on the top of each cylinder, near the exhaust pipes. The main supply steam pipe is connected centrally at the back of the cylinders. Branch pipes form the connection with each slide valve casing by flanges; the expansion joint being on each side of the T piece, which also forms the stop valve casing. The relief valves are respectively situated to retain their requisitions.

As the examples previously alluded to have been separately compared as to the relative position of the starting gear; it will be remembered that the top and sides of the condensers, also the top or upper side of the

cylinder, have been each adverted to as situations for the details in question. Now, Messrs. Napier and Sons—by the arrangement of the starting gear—duly consider the relative positions of the slide valve and connecting rod. They also recognize the requisition of a correct proportion for the length of the eccentric rods to the stroke of the valve; and in the example now under description, the side of the valve casing is deemed the more preferable place for the starting gear.

The type of link adopted is the solid kind, the eccentric rods being connected at the extremities. The slide valve rod is guided by a projection secured in front of the casing; a return connection with the link being effected by a rod, the extremities of which are attached to each requisite detail. The mode of raising and lowering the link is both novel and simple. The acquisition is thus attained: below the link, at the base line, is secured a toothed quadrant, motion being imparted to the latter by a pinion secured on the starting wheel shaft. A lever on the quadrant shaft is connected to the rod which imparts motion—from the link—to the valve rod; the lower eccentric rod being forked for a given length suitably for the requisite connection. By this description it will be understood that the lifting rod is not in this case connected to the link, but to the motion rod connecting the link and slide valve rod. This arrangement admits the advantages of a solid link, combining a comparatively slight adverse motion, which latter attainment can only be appreciated by those who are conversant with the cause and effect of different link motions. The remaining portion of the arrangement under notice is not strikingly novel: it can be truthfully said, however, that originality of design, embracing practical requisitions, has been justly pourtrayed throughout.

SLIPPER GUIDE.

RETURN CONNECTING ROD ENGINE.

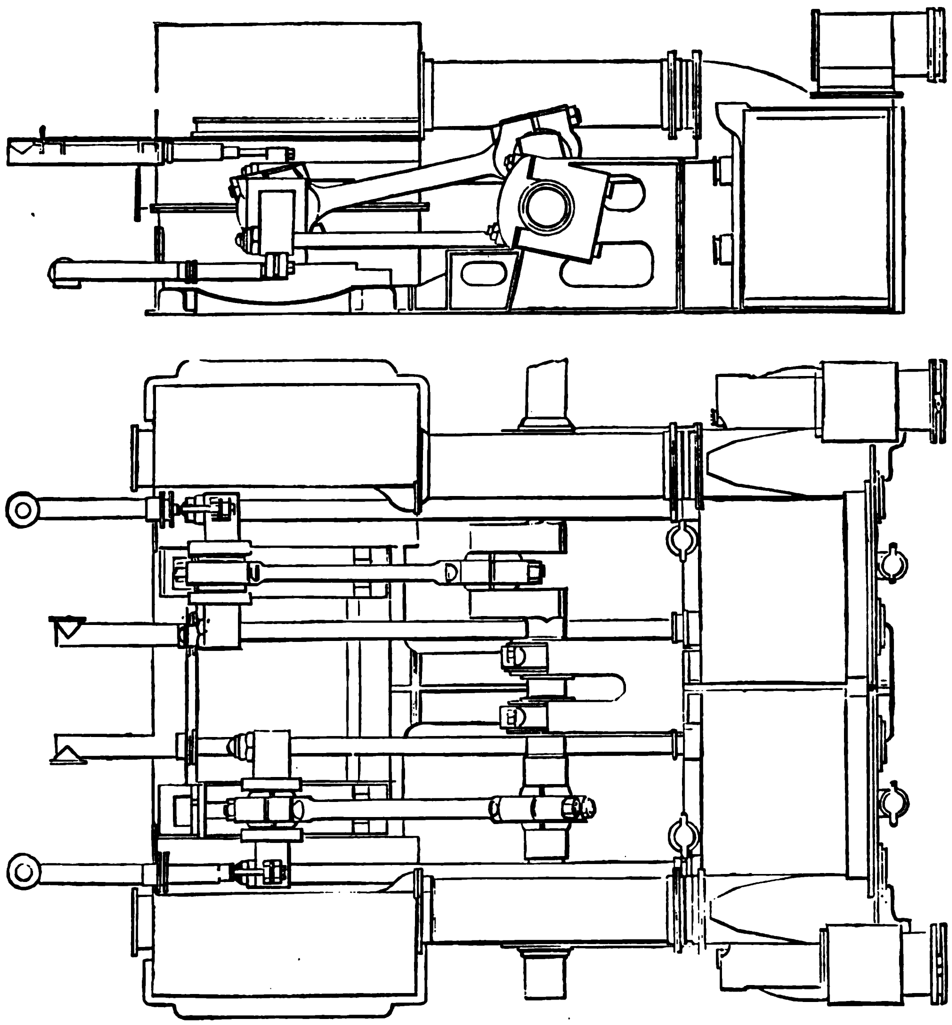
It will have been noticed, in the descriptive and illustrative matter previously given, that the condenser and air pumps for the double trunk and direct acting engines are arranged alike; also the same may be said of those for the return action types. The free space beyond the crank shaft, in the former case, admits of uncontrollable arrangement combining compactness with accessibility. With the return action engine, the piston and connecting rods *cut up* the arrangement of the condenser, the air pumps, however, not being locally affected. Now the space between the suction and delivery valves should always be minimum if possible, remembering the actual requirements in the case of repair, &c. Again, consideration should be given to the motion of the vessel. The surface of the water in the condenser is not always on the horizontal line; hence, with widely disposed arrangements, the air pump is sparely supplied at one portion of the stroke, and as disproportionately flooded at the remainder; and the effect of this inequality of the action is sensibly dispersed throughout the arrangement. The india-rubber valves are greatly strained; also the pump rod and piston packing. The vacuum is impaired: thus a given loss of motive power, to say nothing of the effect of the shock to the parts in working contact, and wear and tear.

The advantage of compact position of details can be appreciated by the mind of ordinary capacity; but the attainment of the same can only be produced by those practically acquainted with the requirements.

The gain by the adoption of separate condensers and pumps is obvious—when the effect of a break down at sea occurs—although, primarily, an increased weight of material and workmanship are involved, proportionately to that for the connected kind. The practical considerations in either case are: access for repair and the retention of a good vacuum; not forgetting also that commercial pause—the first cost, or outlay of speculative capital.

The access, for the purpose of adjusting the guide blocks and channels to the present, have been depicted as being at the top and fore and aft of the arrangement. The firm of Messrs. Maudslay, Sons, and Field—with their double cylinder types and return action—adopt separate condensers and air pumps. By this arrangement, a space centrally between the inner piston rods of each engine is available

Fig. 9.



MESSRS. MAUDSLAY'S MODERN RETURN CONNECTING ROD SCREW ENGINES.

for the purpose of adjustment. The plan and side elevation of the arrangement under notice is represented by Fig. 9—page 63. The following description will render the advantages to be duly appreciated:—

The cylinders are secured together opposite the condensers, each pair being situated at the port and starboard sides of the keel of the hull. The slide valves are of the equilibrium type, double ported, the packing at the back being of the ordinary kind, and a communication from the condenser assists to reduce the face friction. The supply steam enters the slide casing through that for the expansion valve, the latter casing being on the top of the former. Each casing is separately supplied with steam, so that an independent action is preserved.

The link motion next claims attention. In order to preserve a direct action from the eccentric, the link—of the slotted kind—is arranged to rest on the block pin when the link is lowered. With slide valves of maximum area, two rods are introduced, each of which are fixed by nuts to a crosshead; and bracket bushes, secured to the main frame, form guides, through which the slide rods pass. The link block pin is secured to the crosshead centrally at the back, to retain a certain length of eccentric rod.

To correctly understand the arrangement of the gear for raising and lowering the link, a side elevation of it must be imagined. Below the link—centrally of the length of the main frames—is a weigh shaft. On this latter is fixed a double lever: to which at one extremity is hung a counter-balance, and the other is connected to a vertical rod. This rod is attached at the upper end to a coarsely pitched screwed rod, which passes through or fits in a bush supported in a standard, the latter being secured on the slide valve casing. The bush in question forms part of a mitre pinion, which latter gears with another pinion fixed on the hand wheel shaft. Now, it can be readily understood, that on motion being imparted to the mitre gearing, the double lever—by its connection with the screw—will be raised and lowered. The connection of the lever with the link is by a rod, the latter being attached to the centre of the length of the link at the one end, and to the lever by a slot and pin at the other. By this connection an almost equal action is imparted to the slide valve, whether the link is raised or lowered. The position of the starting platform is between the condensers, each engine having separate starting gear.

The hand wheels are situated directly over the connecting rods, about midway of the length of the guide channels; the other manipulating gear being close at hand.

The main frames are designed with a proper attention to the requirements of the same. The adjustment of the main brasses is attained by caps—at the front,—bolts, and nuts. It will be noticed that the centre frame has a double bearing, the central space being provided for the spur wheel which imparts motion to the expansion valves. The crank shaft is in one forging, presenting little for further comment. The connecting rods are singly connected at each end with semi-solid heads and caps, adjusted by securing bolts and nuts; suitable oil cans and wipers for lubrication being correctly situated. The mode of guiding

the piston rods adopted by the firm now alluded to, displays novelty, combined with simplicity. To correctly define, or rather to understand the arrangement, a section at right angles with the line of motion must be presumed. The piston rods, being imperatively situated, greatly affect the form of the crosshead, which in this case is a zigzag form in end elevation, the extremities being connected to the piston rods. The centre portion of the crosshead is horizontal, and turned to form the connecting rod bearing. The guide block —when seen in end elevation—presents the shape of the letter U, the vertical portions of which clasp the crosshead. The section of the guide portion is angular at the sides, and a level surface at the base line; similar, in fact, to the section of the arm of a shaping machine. The adjustment is attained thus: the portion in contact at the base of the channel is separate, the extremities being raised to prevent longitudinal disturbance; and studs inserted in the upper portion, acting on the lower, render the contact of the wearing faces certain. The space between the inside piston rods of each engine admits of free access to the guides, main shaft, bearings, &c.

The feed and bilge pumps receive their motion from projections formed on the crosshead in a line with the piston rods, above and below the same.

The position of the condensers having been alluded to, it next becomes necessary to describe their internal arrangement, and in order to clearly understand the same, a transverse section must be imagined. The air pumps—double acting, receiving their motion direct from the steam pistons—are at the base line. The condensing chamber, for each cylinder, is at the outside of the guide channel. The condensed steam enters the condenser at the front end, the injection entering at the back; a perforated pipe, within the condenser, distributes the water. The suction valves are secured in the floor of the chamber, inverted in action, for the purpose of draining. The bottom of this chamber is on a level with the top of the barrel of the pump. The discharge chamber is formed with the condenser — internally — extending midway transversely, and throughout longitudinally; the height being about one third of that of the condenser. The discharge valves are secured, in the bottom of their respective chamber, above the level of those for suction. For engines of large power, the condensing and discharge chambers—always in one casting—are separate from the pump and valvular passage during construction, but the final connection is made perfect by faced joints, studs, and nuts.

The doors for inspection to all the valves—for air, feed, and bilge pumps—are suitably placed, compatible with the requirements. The snifting, blow-through, cylinder-relief, and other valves, are each respectively situated, the handles, screws, and rods, being arranged to ensure perfect manipulation.

The arrangement now under notice will bear strict analytical comparison with the examples previously illustrated and described; whether for originality to produce the many requisitions, or simplicity of construction to attain the same.

To conclude, it can be said that, with all examples of marine engines, the same results

are sought after, *i.e.*, "speed and economy," combined with "power and safety."

TWIN SCREW ENGINES.

The difficulties once before the marine engineer were, chiefly, the production of an arrangement of engines and boilers for paddle propulsion. Having decided that question, to a certain extent, he hardly achieved the same when the single screw presented itself. After various attempts, each more or less a success, he succeeded in producing fair examples for the purpose required. But science was not exhausted or contented, and thus twin screw propulsion advanced on the field of contention. The marine engineer has—with this last mode—many more potent evils to contend with than with those previously overcome, the arrangement of the engines in the hull being particularly alluded to. To clearly define the difficulties will be to exemplify by practical evidence, having recourse also to dimensions already in use. Take for example, the beam of a vessel to be twenty-five feet, fitted with a propeller eleven feet in diameter, driven by a pair of engines of two hundred horse power collectively. The greatest space occupied by the engines—transversely—from the centre of the crank shaft, to equal eight feet eight inches on the one side, and six feet on the other—return connecting rod type to be adopted.

Now, assume the same hull, or one of similar beam, to be fitted with twin screws, each eight feet in diameter, the distance between the centres being ten feet five inches, and the aggregate power to be as before. By simple calculation it will be found that the same class of engines—secured directly opposite each other—could not be adopted, simply due to the position of the screw shafts. The difficulty then resolves itself into two facts, either the centres of the shaft must be wider apart, or the pairs of engines must be situated fore and aft of each other.

The consideration next claiming attention is the most correct type, or rather the most available, for the system of propulsion now under notice. The amount of space occupied transversely of the hull by the different types of engine, has been fully treated in pages 56 and 57. It now becomes requisite to consider the application in the present case. It has been proved that, for certain proportions of the distance between the centres of the screw to the beam of the vessel, the engines must be arranged fore and aft of each other. With this arrangement, the space longitudinally of the hull is increased relative to that occupied for single screws. It may next be quoted that this difficulty has been met by the advocacy of single engines to each screw shaft for twin propulsion. Now, to render exemplification of practical utility, in all branches of science, recourse as to the application of natural laws will produce a truthful result.

The circle described by the crank pin of an engine is divided centrally by the line of direct action, *i.e.*, when the piston is at the front or back, top or bottom, of the cylinder,—horizontally or vertically situated,—the centre line of the crank is in a direct line with that of the cylinder. The crank pin, therefore, at these points—unless under the effect of momentum—is termed at rest, or at the dead centres. Now from experience, not only known to

marine engineers, it is appreciated that the single engine, if fitted with a fly wheel, may be adopted, in principle, for screw propulsion. It next then becomes necessary to consider the amount of space available, and the conclusion arrived at will be, that the introduction of the fly wheel in the hull is an impracticability. Arguments have been raised that the blades of the propeller may be considered to produce centrifugal force, but this theory is met with practical contradictory results, due to the effect of the waves on the blade. To correctly understand the action of the sea, will be to remember that the vessel is not in all cases floating horizontally or vertically. In some instances two-thirds of the blades of the screw are operating on the air instead of the sister element. Presuming the crank pin to be on either of the "points of rest," and at the same moment a sea impelled the propeller in an adverse direction—two forces acting in contrary directions will, in all cases of mechanics, produce a concussion and momentary stoppage. The progress of the crank would be arrested at the point where it is powerless, and the action of the steam on the piston would assist to render it stationary. It is obvious that the actual effect of the screw propeller is due to the helical form of the blades. The water being neutral—in principle—the revolving motion of the screw imparts a force to the hull. Now, to reverse the action of the two agents—screw and water—the screw must be dragged or pushed through the fluid, and thus a rotative motion will be produced. This latter effect is the only feasible theory that can be brought forward as evidence in favour of the adoption of the single engine. The application in practice is thus: presume the crank pin to be at rest, and the steam exerting its force against the piston, the impetus given to the hull by the prior propulsion, impels the vessel ahead or astern as the case may be. The water becomes the agent of power, and by its contact with the helical form of the blades, tends to cause the latter to revolve, and thus the crank pin will be started. It may now be said, that this being a fact, the barriers before alluded to are of little or no importance. It becomes requisite to consider, however, whether, the *casual* effect with single engines can be deemed equivalent in power to the *certain* action ensured by the duplicate arrangements.

Now, as to the effect of using steam expansively with single engines. The piston when at the full stroke is exposed to the greatest pressure, and immediately the steam is cut off expansion commences, thus the power is decreased in proportion. The sudden shock given to the piston when at the full stroke, impels the same to the opposite end of the cylinder, where fresh steam is introduced to act as a cushion. Now, the force of the piston is transmitted to all the working portions of the detail, and the dead centres of the crank pin will therefore be mostly affected. The lead given to the slide valve *actually* tends to force the piston against its momentum, and until the dead centres—of the crank pin's path—are overcome, the agent employed for progression will be adverse in action.

The remarks in allusion to single engines to the present, have treated of the points of stoppage; attention will now be given to the best means for overcoming the difficulty, by starting gear, *i.e.*, for turning the crank when fixed

Now, the slide valve in this case is powerless; if moved in either direction the effect of the steam on the crank pin will be alike in principle. It then becomes requisite to impart a circular motion to the crank pin, remembering at the same time that the steam is exerting its utmost power on the piston. A sudden disconnection of the starting gear is therefore imperative, but if not certain in action, a total demolition of the weakest part of the entire arrangement will be the result; the effect of which will not only be disastrous, but discreditable to the marine engineer. It will thus be understood that, for steering purposes, a single engine to each propeller is to be avoided, due to the proofs herein given. It may be added that by connecting the shafts, at the extremities, by cranks at right angles with each other, the principle of double cylinder engines will be attained, but at the expense of the steering acquisition. Uniform action is so well appreciated by those who understand its effect, that to dilate on it will be superfluous. The path of the crank pin when divided into two points of connection, at right angles, must be conducive to uniform motion; in fact, engines having three cranks have been proved to be the best.

The principal gain by the adoption of the twin screw system is the availability for manœuvring the hull, whether in a gale or for war purposes. This is produced by the reverse action of the screws—the one propelling, and the other repelling or stationary—thereby causing the centre of gravity of the hull to become a pivot in principle.

Many authorities have prophesied—long before the actual introduction—as to the advantages to be derived from twin screws. Amongst the most truthful may be mentioned, Mr. Roberts, C.E., late of 10, Adam Street, Adelphi, London, who, in 1852, had a patent granted for "The application of independent and separate engines to each shaft of twin screw ships," advocating its adoption for heavy as well as light tonnages. Another veteran engineer—who actually put in practice his ideas on the subject—G. Rennie, Esq., C.E.—many years ago understood the cause and effect. Captain T. E. Symonds, R.N., also a recognised authority, in an excellent paper "On the construction and propulsion of twin screw vessels," 1864, delivered to the members of the "Institution of Naval Architects," duly observes:—

"It will I think be granted that a war ship, no matter for what purpose she may be intended, is inefficient unless possessing the highest attainable speed and power of manœuvring under all circumstances. She should be able to turn in her own length; the steering power should be as perfect going astern as ahead, the propelling and steering agents being out of danger from rams, shot, &c.; and she should have such a draught of water as would enable her to go in and out of our own harbours and docks without the delay or risk now experienced. I do not for a moment question that many of our screw ships may manœuvre very fairly under favourable circumstances; but I confidently state that there is nothing approaching certainty or rapidity in the evolutions of vessels fitted with a single screw. On this point at least I agree with the Chief Constructor of the Navy, who, in his Paper of 1863, states that "our existing ships are incapable of turning 'rapidly.'" Notwith-

standing the position taken by advocates of that system, even they admit the necessity for turning astern till the ship gathers slight sternway before she can be got round; whereas twin-screw ships are turned either way from a state of rest, the first stroke of the screw acting instantaneously on the ship. It is one thing to describe a circle in a fair way, and another to pivot a ship on her centre in either direction, regardless of wind, tide, or obstructions which may occur in action, with the same certainty and ease as you may turn a truck on a turntable.

"It was the absence of this power, experienced in active service, that induced me to turn my attention to the twin-screw system, and in that I have found the only reliable means of attaining it, as I have seen proved on many recent occasions. It also embraces many other collateral advantages, to which I shall briefly allude. Perhaps the most valuable of these is, that by this system we obtain a simplicity of action comprehensible by the veriest tyro in screw tactics, by which the most complicated manœuvres may be unerringly performed with ease and rapidity. This I consider to be the first step to that perfection, in manœuvring long ships, which certainly has not been hitherto attained.

"It has been unjustly complained of by the opponents of the system that it is complicated, and that the engines occupy more room than those of a single screw ship. However, even were it so, the manifest advantage of the duplicate arrangement must weigh down such an objection, especially as applied to ships of war, in which so much depends on their manœuvring, and where a slight extension of engine room space is of no moment in those long ships to which the system is so peculiarly applicable. However, this is not a valid objection, as it has been shown that these engines can be arranged in such a manner as to take up the same room as the ordinary screw engines; but if arranged in the most favourable manner, viz., with two cylinders to each shaft, the loss of longitudinal space is amply compensated for by an equivalent in the side coal bunkers.

"If in working off a lee shore it be found desirable to assist the ship by steam power, the lee screw may be worked with advantage, especially when the ship is at a considerable inclination. With the lee screw thus working, a ship would not fall off in the trough of the sea, as is often the case from inefficient steerage power and other causes in single screw ships. This *specialité* of the twin screw system was fully borne out in recent practice on the American coast in the *Far East*, of 1,300 tons, as communicated to me by her commander. Half power may be made available with economy, under canvas, without overrunning the screw, which could not be the case with the single one.

"*Trial of the Flora.*—The *Flora* is an iron vessel of 450 tons, 160 feet in length, 22½ feet in breadth, and 15½ feet in depth, having two independent engines, with a collective nominal power of 120 horses; the screws working under each quarter before the rudder. The diameter of the cylinders is 26 inches, with a stroke of 21 inches, diameter of screws, 7 feet, pitch 14½ feet. There are two tubular boilers working at 30 lb. pressure, and one high pressure working at 50. This high-pressure boiler is intended to be used for producing a

steam blast in the chimney, and to dry the steam (by admixture) from two common boilers. *The first experiment* was made with both engines, going ahead at full speed, and the helm hard over, when the first circle was made in 3 minutes and 14 seconds, the second in 3 minutes and 13 seconds, and the third in 3 minutes and 16 seconds (the ordinary time, with one screw, being about 5 minutes); the diameter of the circles being about three lengths of the ship, and lessening each time. In the second experiment, one engine and screw worked ahead, the other astern, helm hard over; one circle was made in 3 minutes and 30 seconds, and another in 3 minutes and 40 seconds. In making these circles the ship turned on her centre; the circle was then made the reverse way with the same result. The vessel was then put on a straight course, stopped, and from a state of rest the engines were started, one ahead the other astern, helm amidships; the circle being completed in 3 minutes and 55 seconds, the diameter of the circle being as before within the ship's length. The *Flora*, on her passage over to America, worked one screw, one engine and boiler, on alternate days, making an average speed of 8 knots."

The Messrs. J. & W. Dudgeon, Marine Engineers and Architects, have had much experience in the matter under notice, having constructed many vessels and engines on the twin-screw system. These productions have given good results, embracing economy, speed, and the superlative steering advantages attained by the screws working separately.

In a record of performance and experiences with twin screw steamers, built by the firm quoted—prepared for the Institution of Naval Architects—these gentlemen state :—

"Since March, 1862, we have completed twenty twin-screw steamers, of varying tonnage and power, up to 1,500 tons and 350-horse power. First, we built eight vessels of 425 tons and 120-horse power, all destined for blockade running. They have all been very successful in that hazardous trade, some running as many as a dozen times, the average number of runs being five. At their trial trips they showed the following results: —with an indicated horse power = 600 (attained by four cylinders 26 inches diameter by 21 inches stroke; steam, 23½ lbs.; vacuum, 26 inches; revolutions, 115). The vessels went 14 knots; the immersed midship section was then 150 square feet, and the displacement 400 tons; the consumption 15 cwt. per hour. All these numbers show the mean of the eight respective trials of these vessels, which, being sister ships, were on these occasions trimmed nearly all alike, and tried under equal circumstances. As regards manœuvring the ship, the following was observed :—

"1. Full speed ahead with both engines, helm put hard over, the circle was completed in three times the ship's length in 3 minutes 33 seconds.

"2. One engine going full speed, the other stopped, helm hard over on working side, the circle was completed in a little more than ship's length in 3 minutes 30 seconds.

"3. One engine going ahead, the other astern, helm hard over, the vessel turned on the centre of her length completely round in 3 minutes 48 seconds.

4. The engines going in opposite directions,

as before, but helm fixed amidship, the circle was made again in the ship's own length in 4 minutes 28 seconds.

"The facility of manœuvring has been of very great service to these vessels in performing their duty. The fact is, it has saved them in many cases from being taken. The advantage of having two engines independent from each other has also manifested itself; when one engine was damaged, it was then stopped and easily repaired, whilst the other was going. Also, in going out, it was found saving to work alternately one engine and boiler, when the vessel still attained the fair speed of eight knots.

"The *Edith* was a blockade runner of 531 tons and 200-horse power. On her trial trip she went 13·4 knots, with an indicated horse power = 894 (attained by two pairs of 34-inch cylinders, by 21-inch stroke; steam, 21 lbs.; vacuum 26½ inches; revolutions, 108). The immersed midship section was then 180 square feet, the displacement 510 tons, and the consumption 24 cwt. per hour. In manœuvring the ship by her screws we found—

"1. Both engines going ahead, helm hard over, the full circle was made in 4 minutes 3 seconds.

"2. Screws going opposite ways, the circle was done in 3 minutes 29 seconds. The circle being completed, the engines were suddenly reversed to opposite directions, and their action on the vessel was instantaneous, the revolving motion of the ship being checked and reversed with the greatest ease. This experiment was repeated several times, and proved that such a twin screw vessel might in itself become the carriage for very heavy ordnance, too heavy to be trained in the ordinary way.

"3. With one engine going, the other stopped, helm hard over, the full circle was made in 4 minutes 31 seconds."

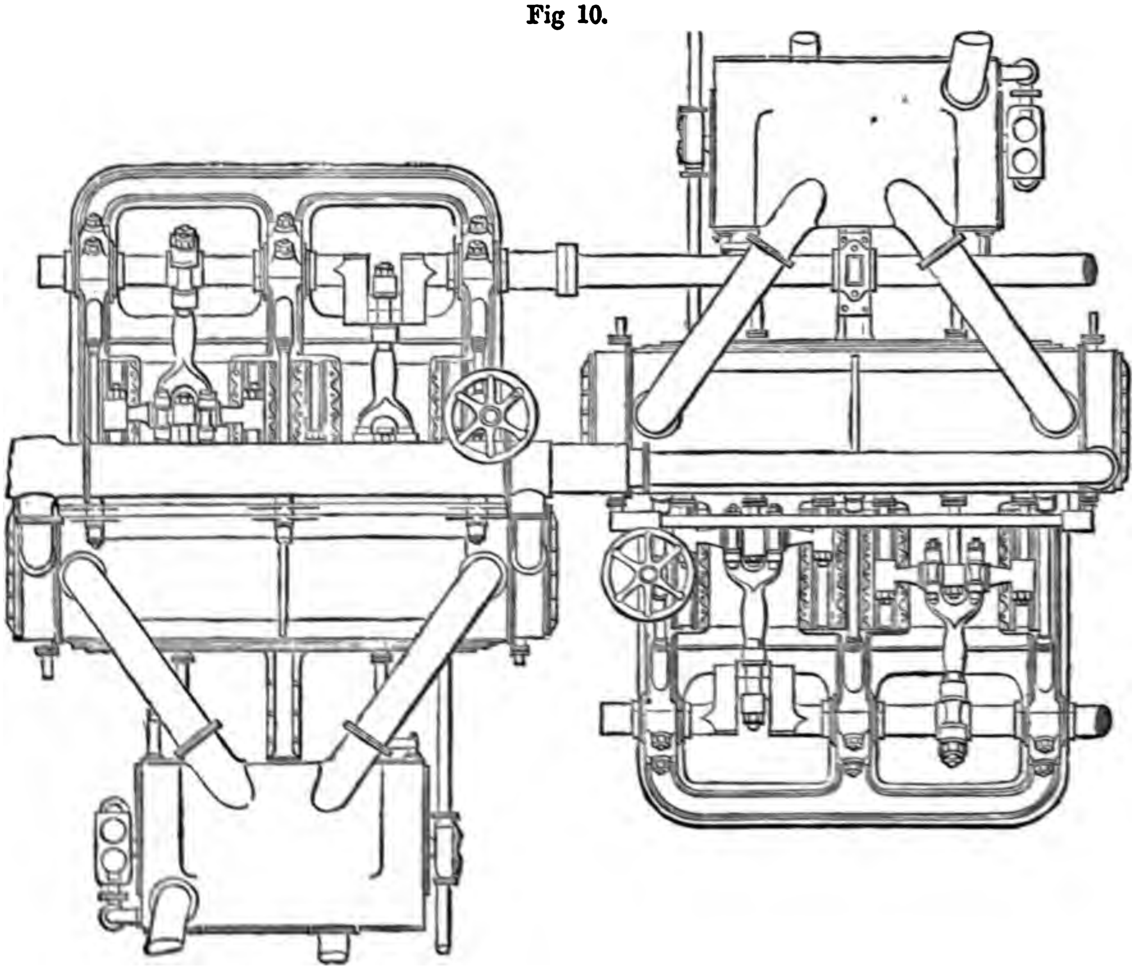
These quotations from reliable authorities develop the fact that the ready manipulation of the engine is the main consideration with the system advocated.

The position of the starting gear will greatly depend on the type of engine adopted. The observance of the two facts, "time and power," in the present case—alike with the single screw—decides the arrangement of the gear. The pair of engines to each screw, that can be separately regulated, started, stopped, and reversed by one engineer, must not be compared with those requiring individual manipulation. The starting wheels that are horizontally supported, with vertical handles singly to each wheel, must not be confused with those requireing a vertical position, with a plurality of handles to each rim. To determine the best position for the starting wheel will be to remember the requirements in each case. With inverted and horizontal arrangements, oppositely placed, the fore part of the engines may be considered the better situation. In the case of the engines being arranged fore and aft of each other, a central position for the wheel in question is doubtless the most convenient: it may be added that these various positions are quoted from practical examples.

The illustration, Fig. 10—page 72—represents the plan of an arrangement of single piston rod direct acting engines by the firm of Messrs. J. & W. Dudgeon. Each arrangement is separate—being, in fact, duplicates, without any connection of the working por-

tions. The cylinders are the compound kind: the high pressure within the low. Three piston rods are required for each engine, the centre rod being connected to the high pressure piston. The guide channels—below each piston rod, connected to the annular piston—are the ordinary kind, arranged to receive slipper blocks. at the back end to the steam cylinders. The brasses for the main bearings are adjusted, at an angle, by the ordinary securing bolts and caps. The slide valves are double ported, to supply and exhaust the steam simultaneously from the respective cylinders, one valve only being requisite to each engine. The larger

Fig 10.



MESSRS. DUDGEON'S COMPOUND TWIN SCREW ENGINES.

The crosshead is secured to the piston rod by nuts, and turned on each side of the central connection to receive the forked end of the connecting rod. This latter detail is the T end kind, with flat brasses, caps, and securing bolts. The main frame, crank shaft supports, and the connecting portions are in one casting, secured cylinder is double ported on each side of the exhaust port, and the requisite passages—at the back and front ends—communicate with the high pressure cylinder.

The link-motion presents much evidence of care as to its correct action, the arrangement being thus—The link is two solid bars,

connected transversely at the extremities. The block is inserted in a single eye, which latter forms the extremity of the valve rod. Each bar of the link clasps the block, suitable recesses being formed. The eccentric rods are connected to the outsides of each bar, or clasp the same, secured by separate pins. The lifting rod is connected, at the lower end, to the centre of the length of the link; the upper end being attached to a lever, whose shaft is supported in brackets, the latter being secured to the front of the cylinder. Motion is imparted to the lever weigh shaft by a toothed quadrant keyed thereon. The hand wheels are horizontally secured on vertical shafts; the latter are formed with worms, which gear with the quadrants alluded to. The wheels in question are situated at the inner extremities of each main framing. The starting platform, above the steam piping, supports the wheel shaft columns, thus combining simplicity of connection with economy of construction.

The condensers—seen beyond the cylinders, at the back of the same—are the surface kind. The motion requisite for the air and circulating pumps—double acting—is derived from the steam pistons. The exhaust steam enters the condenser at the top by separate pipes, connecting with the exhaust passages on the cylinders. The supply steam pipe is placed almost central of the entire arrangement, passing from end to end, and thus being common to each slide casing. Suitable branch pipes and stop valves are correctly arranged, according to their respective requirements.

The feed and bilge pumps are worked by arms secured on each air pump rod. The requisite valve boxes are secured at the sides of each condenser, to ensure access for inspection and repair. The remaining portion of the requisitions, imperative with all marine engines, are respectively situated; both access and ready manipulation having met with due consideration.

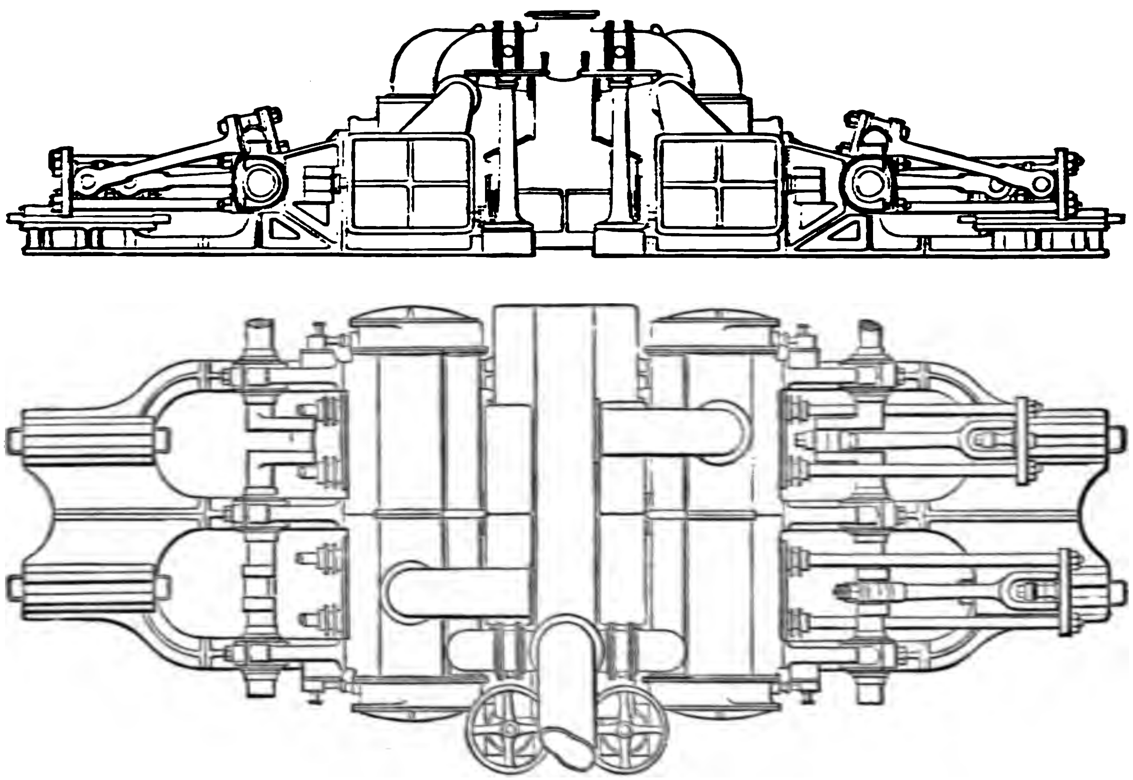
The return connecting rod engine is particularly adapted for the twin screw system, on account of their occupying less room transversely of the hull than any other type.

In arranging these engines in the hull, prior consideration should be given to the access for repair and adjustment, remembering also that the space occupied by engines of minor power is not in consecutive proportion with those of large size. As before alluded to, the position —or distance between the centres—of the screw shafts will in all cases determine the position of the engines. With the return action type, the space occupied beyond the crank shaft, at the cylinder end, will be known by—half stroke of piston + half of diameter crank pin, and clearance for head of connecting rod + length of cylinder complete. It will be found that, as a rule—with few exceptions—when the types in question are arranged opposite each other, there will be sufficient space between the cylinders for the requisite inspection and repair. The Messrs. Maudslay, Sons, and Field, arrange their return connecting rod engines for twin screws, as referred to, and by securing the condensers —between the piston rods of each engine— beyond the crank shaft, ample space is allowed where required.

For vessels of light draught and wide beams, the machinery should be distributed in

accordance. The illustration, Fig. 11, on this page, is a plan and side elevation of two pairs of engines designed by ourselves for a Despatch vessel—particular attention being given to compactness of arrangement. The type of engines is the return connecting rod, of the ordinary kind, similar in principle to those already alluded to for the single screw. the back end covers of the fore and aft cylinders. In the event of a fracture, a reverse connection is attainable, blank flanges being suitably provided. The floors of the condensing and discharge chambers are above the top of the pump barrels. The suction valves are inverted, and those for the discharge reverse in action, both sets being on

Fig. 11.



BURGH'S TWIN-SCREW RETURN-ACTION ENGINES.

the same level. The final discharge water pipe can be secured either at the aft end of the chamber, or the top of the same; the former being preferred in this case.

Ample space being available in this case, the condensers are situated between the back ends of the cylinders.

The condensing chambers are fore and aft, or on each side of the discharge chamber, which is common to both air pumps. The motion requisite for the pumps—double acting one to each pair of cylinders—is imparted from the steam pistons by rods passing through

The steam cylinders are provided with exhaust passages, communicating on the top of the peripheries. On the top of the exhaust passages—of the fore and aft, port and starboard cylinders—are secured two pipes, each

being common to the respective cylinders, and connected to the condensing chambers. The supply steam pipes are secured to the slide valve casings, over the back ends of the cylinders. The main supply pipe is connected at the forward end of the arrangement.

The slide valves are the equilibrium double ported type, similar to those already described. The link motion also presents no novelty for further comment. The hand wheels and shafts are supported in vertical columns—seen in plan and elevation. The feed and bilge pumps are secured at the sides of the frames, motion being derived from the piston rods, or direct from the steam pistons, if preferred. The doors for access to the valves of both air, feed, and bilge pumps are respectively arranged; also the supplementary valves, detailed allusion to which, for the present purpose, is not requisite. It may be added, that in the event of the space between the back ends of the cylinders not being sufficient for the condenser, &c., as at present arranged, the arrangement adopted by the Messrs. Maudslay will be the better.

Another mode of arranging the cylinders and condensers—when oppositely situated—is to secure the condensers between the crank shafts, and the cylinders beyond the same, at each side of the hull. Some authorities prefer the screws as close together as their diameters will permit, for protection, or prevention from fouling. Other advocates recognize the steering acquisition only, and thus increase the space between the screw shafts proportionately. In conclusion, therefore, it can be stated that the beam of the vessel regulates the type of engine for twin screw propulsion.

CHAPTER III.

ARRANGEMENT OF BOILERS, AND COMBUSTION.

The prior considerations necessary in designing machinery are: *material, mode of construction, form, and repair*, and the best means of general access. A notice of these four requisitions is imperative when designing Marine Engines, and similarly with Boilers, but the application in each case widely differs. In order to render this fact obvious, a brief comparative allusion to each will be desirable.

Material.—The principal detail of an engine, exposed to the steam, is the cylinder, formed of cast-iron, in one mass. The boiler is of wrought-iron, in many portions—it being remembered that in each case the strains imposed must be duly considered.

Mode of Construction.—The cylinder requires pattern-making, moulding, boring, facing, and fitting, each portion being separate trades. The manufacture of the boiler consists of forging, plate bending, riveting, staying, and caulking, each being also separate crafts of workmanship.

Form.—The cylinder, from its name, is understood to be simple in shape to mould, while at the same time the greatest strength is attained—globular excepted. The boiler is subservient to the hull for which it is destined, therefore no rule as to form can be given. A high boiler can be of given proportions as to height, breadth, and length, which for a low boiler are entirely altered; the tubes in the high boilers being over the fire boxes, while for the low example they are at the side.

Repair.—This natural effect is the main cause for the schemes and arrangements so often introduced—only to be discarded. The engineer is, perhaps, dissatisfied with a tried example, and starts a novelty, which, while in working order, produces better results than the original plan. But directly repair or cleaning is required, it is too often proved that a complicated arrangement is foreign to accessibility or economy. The steam cylinder is a fair example of mechanism for comparison; for example, the single piston rod type is the most accessible for internal inspection or repair. The compound arrangement was introduced with the view to economize the steam, but complication and inaccessibility for adjustment were likewise imperative. The compound engine cylinders are, as before stated, often secured together in a line with each other, the central piston rod connecting the pistons. It is therefore obvious that the intermediate stuffing box is inaccessible, without disarranging the larger or lesser pistons.

Now as to the impurities entering the cylinders, they may be considered to be three kinds: steam, water, and oil or tallow. The first of these essentialities proceeds from the boiler,

hence the chemical properties of the water therein, and the amount of steam space proportionately, will greatly regulate the nature of the steam. The presence of water in the cylinder is dreaded by all engineers, who are cognizant of its cause and effect. The cause is due to priming, or violent ebullition in the boiler, while the effect reduces the pressure of the steam, and injures the surfaces in working contact. It may be added in passing that a full investigation of the cause and prevention of priming can be found under its particular heading.

The tallow or oil in the cylinder is for the purpose of lubrication, or preserving the polished frictional surfaces. When highly super-heated steam is used, the unguent is destroyed, or rather its powers of lubrication, and by becoming hard and dry, *cuts* the surface of the cylinder facings. Evidence of these facts are plentiful; in some instances grooves one-sixteenth to three-eighths of an inch deep have been formed in a few hours.

The repairing attainments for the cylinders consist of re-boring and planing, or facing the respective parts, which restore the same to their original state. The boiler, when worn thin, or over-strained, must be patched in the one case and additionally stayed in the other. The access for repair also is cramped, requiring that the workman should assume the most uncomfortable attitudes: this fact can be actually understood only by those who have had practical experience in the matter under comment. The facilities for repairing the cylinder and the boiler are obviously in favour of the former. The causes for repair, also, in the cylinder, are less than those in the boiler. The latter contains two of the evils presented in the former, together with two others, which the cylinder escapes, viz., *combustion*, with combination, or complication, of construction, and the retention of the solid impurities in the water, termed *scale*.

It will be remembered that in the "prefatorial remarks" a brief allusion was made to "heat and combustion;" it is now intended to further probe these subjects in a more scientific as well as practical form.

The general intention of the designer of the marine boiler is to attain evaporation within as small a compass as practicable. Now it is obvious that combustion produces heat, but the most economical mode of attaining the same is the object to be gained.

Commencing descriptively with combustion, it will not be out of place to entertain the opinions of recognised authorities. Professor Rankine, in his work on the "Steam Engine and other prime movers," duly observes:—

"Every chemical combination is accompanied by a production of heat: every decomposition by a disappearance of heat, equal in amount to that which is produced by the combination of the elements which are separated. When a complex chemical action takes place, in which various combinations and decompositions occur simultaneously, the heat obtained is the excess of the heat produced by the combinations above the heat which disappears in consequence of the decompositions. Sometimes, also, the heat produced is subject to a further deduction, on account of heat which disappears in melting or evaporating some of the substances which combine, either before or during the act of combination.

"*Combustion* or *burning* is a rapid chemical combination. The only kind of combustion which is used to produce heat for driving heat engines, is the combination of fuel of different kinds with oxygen. In the ordinary sense of the word *combustible*, it means, *capable of combining rapidly with oxygen so as to produce heat rapidly*. By an *elementary* or *simple substance* is meant one which has never been decomposed.

"The chief elementary combustible constituents of ordinary fuel are *carbon* and *hydrogen*. *Sulphur* is another combustible constituent of ordinary fuel; but its quantity and its heat-producing power are so small, that its presence is of no appreciable value.

"The ingredients of every kind of fuel commonly used may be thus classed—

"(I.) *Fixed or free carbon*, which is left in the form of charcoal or coke after the volatile ingredients of the fuel have been distilled away. This ingredient burns either wholly in the solid state, or part in the solid state and part in the gaseous state, the latter part being first dissolved by previously formed carbonic acid, as already explained.

"(II.) *Hydrocarbons*, such as olefiant gas, pitch, tar, naphtha, &c., all of which must pass into the gaseous state before being burned.

"If mixed on their first issuing from amongst the burning carbon with a large quantity of air, these inflammable gases are completely burned with a transparent blue flame, producing carbonic acid and steam. When raised to a red heat, or thereabouts, before being mixed with a sufficient quantity of air for perfect combustion, they disengage carbon in fine powder, and pass to the condition, partly of marsh gas, and partly of free hydrogen; and the higher the temperature, the greater is the proportion of carbon thus disengaged.

"If the disengaged carbon is cooled below the temperature of ignition before coming in contact with oxygen, it constitutes, while floating in the gas, SMOKE, and when deposited on solid bodies, SOOT.

"But if the disengaged carbon is maintained at the temperature of ignition, and supplied with oxygen sufficient for its combustion, it burns while floating in the inflammable gas, and forms RED, YELLOW, or WHITE FLAME. The flame from fuel is the larger, the more slowly its combustion is effected.

"(III.) *Oxygen and hydrogen* either actually forming water, or existing in combination with the other constituents in the proportions which form water. According to a principle already stated, such quantities of oxygen and hydrogen are to be left out of account in determining the heat generated by the combustion. If the quantity of water actually or virtually present in each pound of fuel is so great as to make its latent heat of evaporation worth considering, that heat is to be deducted from the total heat of combustion of the fuel.

"The presence of water, or its constituents, in fuel, promotes the formation of smoke, or of the carbonaceous flame, which is ignited smoke, as the case may be, probably by mechanically sweeping along fine particles of carbon.

"(IV.) *Nitrogen*, either free or in combination with other constituents. This substance is simply inert.

"(V.) *Sulphuret of iron*, which exists in coal, and is detrimental, as tending to cause spontaneous combustion.

"(VI.) *Other mineral compounds* of various kinds, which are also inert, and form the ASH left after complete combustion of the fuel, and also the *clinker*, or glassy material produced by fusion of the ash, which tends to choke the grate."

These laws of combustion are of course founded on natural causes, remembering also that each kind of fuel generates gases of respective proportions. The ratio of the carbon to the hydrocarbons will not in all cases be alike; much depends on the class of fuel used. These may be taken—for the present purpose—to be of two independent kinds—coal and wood. Their classification and chemical properties are thus arranged by Professor Rankine :—

"*Coal.*—The extreme differences in the chemical composition and properties of different kinds of coal are very great; but the number of those kinds is very great, and the gradations of their differences are small.

"The proportion of free carbon in coal ranges from 30 to 93 per cent.; that of hydrocarbons of various kinds from 5 to 58 per cent.; that of water, or oxygen and hydrogen in the proportions which form water, from an inappreciably small quantity to 27 per cent.; that of ash, from 1½ to 26 per cent.

"The numerous varieties of coal may be divided into principal classes, as follows :—

"Anthracite or blind coal.
"Dry bituminous coal.
"Caking coal.
"Long flaming or cannel coal.
"Lignite or brown coal.

"*Anthracite* or *blind coal* consists almost entirely of free carbon. It has a colour intermediate between jet black and the greyish-black of plumbago, and a lustre approaching to metallic.

"Its specific gravity is from 1·4 to 1·6, that of water being 1.

"It burns without smoke, and, when dry, without flame also; but the presence of moisture in it produces small yellowish flames.

"It requires a high temperature, and in general a blast produced by mechanism, for its combustion. If suddenly heated, it splits into small pieces, which are liable to fall through the grate bars of the furnace and be lost. In furnaces where it is used, therefore, each fresh portion should be gradually heated before being ignited.

"*Dry bituminous coal* contains on an average from 70 to 80 per cent. of free carbon, about 5 per cent. of hydrogen, and 4 per cent. of oxygen; so that 5½ per cent. of hydrogen is available to produce heat. This hydrogen exists in combination with part of the carbon. Such coal burns with a moderate amount of flame, and little or no smoke. Its average specific gravity is about 1·3.

"*Bituminous caking coal* contains on an average from 50 to 60 per cent. of free carbon, and about equal weights of hydrogen and oxygen, amounting to from 10 to 12 per cent. of its weight. It softens when exposed to heat, and pieces of it adhere together. It produces more flame than dry bituminous coal, and also produces smoke, unless that is prevented by special means. Its average specific gravity is about 1·25.

"*Long flaming coal* differs from the last variety chiefly in containing more oxygen. In some examples it softens and cakes in the fire; in others not. It requires special means for the prevention of smoke.

"*Brown coal,* or *lignite,* is found in more recent strata than any of the preceding kinds. It is intermediate in appearance and properties between them and peat. It contains on an average from 27 to 50 per cent. of free carbon, about 5 per cent. of hydrogen, and 20 per cent. of oxygen. Its specific gravity is from 1·20 to 1·25.

"With respect to the different kinds of coal, M. Peclet makes a remark to the effect, that the caking bituminous coals pass to the dry coals and to anthracite by diminution of their oxygen and hydrogen, and to the long flaming coals and lignites by the augmentation of their oxygen.

"From the specific gravities already stated, it appears that a cubic foot of solid coal weighs from 70 to 90 lbs.; but coal in pieces, such as are commonly used for feeding furnaces, including the spaces between the pieces, occupies from $1\frac{3}{10}$ to $1\frac{4}{10}$ times the space that the same coal fills in a continuous mass; so that the average weight of coals, including the space between the pieces, is about 52 lbs. per cubic foot. In a few examples it is as high as 56 or 60 lbs. to the cubic foot.

"*Wood,* when newly felled, contains a proportion of moisture which varies very much in different kinds and in different specimens, ranging between 30 and 50 per cent., and being on an average about 40 per cent. After eight or twelve months' ordinary drying in the air, the proportion of moisture is from 20 to 25 per cent. This degree of dryness, or almost perfect dryness if required, can be produced by a few days' drying in an oven supplied with air at about 240° Fahrenheit. When coal or coke is used as the fuel for that oven, 1 lb. of fuel suffices to expel about 3 lbs. of moisture from the wood. This is the result of experiments on a large scale by Mr. J. R. Napier. If air-dried wood were used as fuel for the oven, from 2 to $2\frac{1}{2}$ lbs. of wood would probably be required to produce the same effect.

"The specific gravity of different kinds of wood ranges from 0·3 to 1·2.

"*Perfectly dry* wood contains about 50 per cent. of carbon, the remainder consisting almost entirely of oxygen and hydrogen in the proportions which form water. The coniferous family contain a small quantity of turpentine, which is a hydrocarbon. The proportion of ash in wood is from 1 to 5 per cent. The total heat of combustion of all kinds of wood, when dry, is almost exactly the same, and is that due to the 50 per cent. of carbon."

The chemical properties of each kind of fuel used in marine boilers can thus be readily understood. It is next necessary to consider that best adapted for evaporation, remembering the most attainable at the same time. From experience it is known that furnaces intended to burn wood and coal must be of different proportions. The quantity of heat, given out by a given supply of coal, will be more in proportion to that produced by the same cubical amount of wood. This is due to the quantity and nature of the gases in each kind of fuel adopted. The flame from the vegetable production is generally purer, but giving out less proportionately than that from the mineral kingdom. It is well known that the less colour in the flame the greater the heat emitted from the same, the electric and lime-light for example. Now with the ordinary coal, the flame is a deep yellow—finally almost white—a red colour producing the most smoke.

This latter contains the least heat of all the compounds, being formed of minute particles, but not entirely destitute of independent combustive power: evidence of this is seen with ignited chimneys, &c.

The ordinary rate of combustion—with coals—in marine boilers is assumed at *sixteen* to *twenty-five* pounds per hour to each square foot of grate surface. The amount of fuel that *can* be burnt *economically* in a given time, depends on many causes. The state of a fire at the commencement—say the first hour or two—differs with that in double the time. Clinkers, in a molten and solid state, become formed, and thus the draught *under* the fire bars is impeded. The advantage of a scientific mode of stoking is only appreciated by those who are acquainted with the same.

The correct amount of air or draught required will be entirely regulated by the cubic contents of the furnace to the area of the grate surface. Indeed it may be said that ample facilities for increasing or decreasing the action of the air, both above and below the fire, should always engage prior attention when designing marine boilers.

Professor Rankine states:—

"The number of pounds of air required in order to supply the oxygen necessary for the combustion of one pound of any sort of fuel whose chemical composition is known.

"To express that weight symbolically, let it be denoted by A.

"Then C. H. and O. having the same meanings as before.

$$A = 12\,C + 36\,(H \tfrac{O}{8}).$$

"The following are a few of the results:—

Fuel.	Carbon. C.	Hydrogen. H.	Oxygen. O.	Weight. A.
Coal—anthracite . .	0·915	0·035	0·026	12·13
„ dry bituminous .	0·87	0·05	0·04	12·06
„ caking	0·85	0·05	0·06	11·73
„ „	0·75	0·05	0·05	10·58
„ cannel	0·84	0·06	0·08	11·88
„ dry long flaming	0·77	0·05	0·15	10·32
„ lignite	0·70	0·05	0·20	9·30
Wood—dry	0·50	. .	. .	6·00

"It is unnecessary for practical purposes to compute the air required for the combustion of fuel to a great degree of exactness; and no material error is produced if the air required for the combustion of every kind of *coal* and *coke* used for furnaces is estimated at *twelve pounds per pound of fuel.*

"Besides the air required to furnish the oxygen necessary for the complete combustion of the fuel, it is also necessary to furnish an additional quantity of air for the *dilution* of the gaseous products of combustion, which would otherwise prevent the free access of air to the fuel.

"The more minute the division, and the greater the velocity with which the air rushes amongst the fuel, the smaller is

the additional quantity of air required for dilution.

"From the various experiments, especially those made for the American government by Mr. Johnson, it appears that in ordinary boiler furnaces, where the draught is produced by means of a chimney, the weight of air required for dilution is equal to that required for combustion; so that if A′ denotes the total weight of air to be supplied to the furnace per lb. of fuel,

$$A' = 2\,A = 24 \text{ lbs. nearly.}$$

"But in furnaces where the draught is produced by means of a blast pipe, like those of locomotive engines, or by means of a fan, the quantity of air required for dilution, although it has not yet been exactly ascertained, is certainly much less than that which is required in furnaces with chimney draughts; and there is reason to believe that on an average it may be estimated at about *one-half* of the air required for combustion; so that in this case,

$$A' = \frac{3}{2}\,A = 18 \text{ lbs. nearly.}$$

"This estimate is roughly made, but it is the nearest approximation at present attainable. It is probable that the supply of air required for dilution varies considerably in different arrangements of furnace, and for different kinds of fuel; and it is possible, that by blowing the air for combustion into a furnace in small enough jets, and with sufficient force, air for dilution might be rendered unnecessary, so that A′ would be = A.

"An insufficient supply of air causes imperfect combustion of the fuel, which in bituminous coal is indicated by the production of smoke, and in coke and blind coal by the discharge of carbonic oxide gas from the chimney. That gas is transparent and invisible; but its presence may be detected by the blue or purple flame with which it burns when ignited in contact with fresh air.

"An excessive supply of air causes waste of heat to the amount corresponding to the weight of air in excess of that which is necessary, and to the elevation of the temperature at which it is discharged from the chimney above that of the external air.

"In burning charcoal, coke, and coals which contain a small proportion only of hydrocarbons, a supply of air sufficient for complete combustion will enter from the ash pit through the bars of the grate, provided there is a sufficient draught, and that care is taken to distribute the fresh fuel evenly over the fire, and in moderate quantities at a time, so that the thickness of the layer of burning fuel shall never differ much from ten or twelve inches."

Marine engineers generally attach doors in front of the ash pit, and in some cases on a level with the crown of the furnaces, to regulate the quantity of air to be admitted to accelerate combustion. By this arrangement *practice* becomes the best teacher, remembering also that the fuel and mode of stoking must be duly considered.

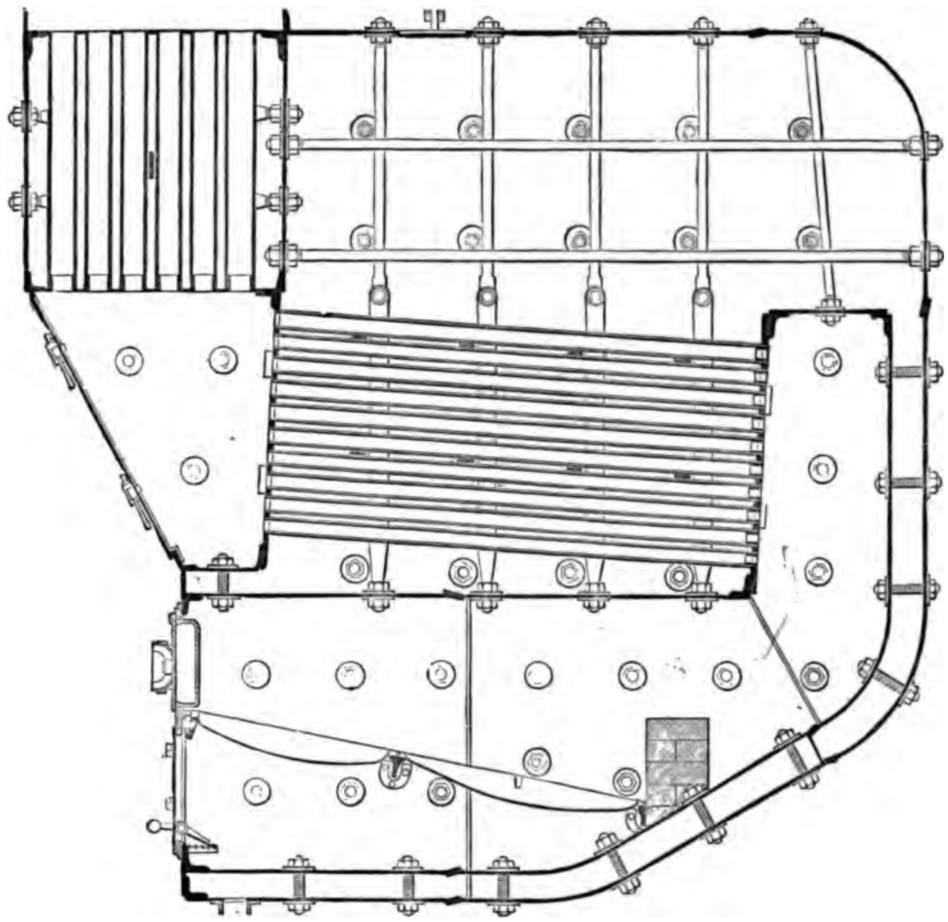
With reference to the actual heat produced by combustion, or rather proportionately to the amount of fuel consumed, much depends on the nature of the fuel and management of the same. Good coal with great combustive powers—viz., of an expanding nature when heated, giving out gas in profusion—should be used in small quantities, of uniform size, to produce the greatest effect. It must be stated,

ɜr, that the coals generally used for purposes are of a second-rate quality, size, partially dust. A great deal of ıt fuel is doubtless often wasted by the ɜness of the stoker. A certain quantity rough the spaces between the fire bars—he fire is clear—and in such cases much overboard with the ashes.

may arise from careless firing; but the amount which is unavoidable with good firing has, in some cases, been ascertained by experiment, and found to range from nothing up to 2½ per cent."

The internal portions of the marine boiler, exposed to the effect of the flame and heated gases, are the fire box, combustion chamber, tubes, and smoke box.

Fig. 12.



SECTIONAL ELEVATION OF A GENERAL MARINE BOILER WITH SUPER-HEATER, AS USED AT PRESENT.

h these facts as certainties, it is almost ible to arrive at a datum that will produce ɜct proportion. Professor Rankine also vledges this by stating:—It is impossible nate the greatest amount of waste which

To enable the student to fully comprehend the future analysis of the theory of the action of the flame on each portion, the above illustration—Fig. 12—is introduced. The boiler represented is the "high type," so called from the

perpendicular space allotted. The fire box and combustion chamber are partially separated by a brick bridge. The chamber extends, in height, above the level of the top row of tubes; this is due to construction rather than for evaporation. The tubes have a slight rake, ascending at the smoke box end, this latter being the final evaporator. The vertical tubes, secured in the upper portion of the smoke box, are for super-heating, to which separate allusion will be made. The fire bars are inclined towards the inner extremities for two purposes: the one to assist the action of stoking or agitating the fire, and the other to accelerate the draught, it being remembered that the greater portion of the air enters below the bars.

Presuming the fuel to have been ignited, and sufficient time to have elapsed to form combustion, the flames and gases will not rise perpendicularly, but will rather form a series of curves, striking against the crown of the fire box. Some authorities prefer to assume that the line of flame contact with the fire box is at an angle; the former, however, is the more truthful conclusion, as practice testifies. The line of the flame's progress after leaving the fire box is directly over the bridge. The use of this latter introduction is for two purposes: first, to prevent the fuel from entering the chamber beyond; and secondly, to increase the effect of the flame on the crown of the fire box. It is obvious therefore, from the causes alluded to, that the flame produces as much evaporation directly over the bridge as at any other place. The bridge also *checks* the flame, and thus allows "time" for combustion. So far is this certain, that inverted bridges have been introduced, producing excellent results. In other examples, the crown of the fire box has been lowered at the inner end, thus decreasing the area for the first passage.

The ignition of the fuel commences almost directly inside the fire door; the actual distance being from three to six inches. Should air be admitted above the grate, the flame will be driven forward in proportion to the force of the *crown* draught. It may be added in passing that many authorities advocate the introduction of air above the grate, while others denounce the same; suitable provisions for regulation, however, should always be introduced. In order to attain the regulation, many schemes have been promulgated, some self-acting by the draught, others regulated by the hand, and, in some cases, the pressure of the steam has been resorted to as the principal agent.

Mr. Fairbairn, in his "Useful Information," when treating of the "temperature of furnaces," informs engineers that:—

"It is a difficulty of no ordinary description to ascertain with sufficient accuracy the temperature of a furnace. In fact, every fire and every furnace is continually changing its temperature, as well as the nature of the volatile products, as they pass off during the process of combustion. When a furnace is charged with a fresh supply of fuel, its temperature is lowered, and that from two causes: first, by the absorption of heat which the coal fuel takes up when thrown upon the fire; and, secondly, by a rush of cold air through the open door of the furnace. Attempts have been made to remedy these evils by the aid of machinery and continuous firing; but taking the whole of the existing schemes into account, and bestowing upon them the most favourable con-

sideration, it is questionable whether they are at all equal (either as regards efficiency or economy) to the usual way of working the fires by hand. I am persuaded the latter plan is the best; and, provided a class of careful men were trained to certain fixed and determined regulations, and paid, not in the ratio of the quantity of coals shovelled on the fire, but in proportion to the saving effected, we should not then have occasion for the aid of machinery as an apology for inattention and ignorance.

"On a cursory view of the subject, it is obvious that the quantity of air necessary to be admitted will greatly depend upon the nature and quality of the fuel used. In a light burning fuel, such as splint and cannel coal, less air will be required, as the charge burns freely with clear spaces between the grate-bars, and is attended by less risk of cementation than the caking coal, which in some cases completely seals the openings, and thus deprives the fuel of that quantity of air necessary for its combustion. Under such circumstances a permanent opening will be found exceedingly efficacious, and that more particularly when the heat vitrifies the earthy particles of the coal, and forms clinkers on the top of the grate bars. In the use of this description of fuel the permanent apertures are of great value.

"The constituents of coal vary in quality as well as degree, and this may be seen from what has already been stated in the analysis given by recognised chemical authorities at the commencement of the inquiry. In order, therefore, to provide for these differences, and to effect its perfect and economical combustion, it will be necessary to show in what manner and under what circumstances its gaseous products combine with other elements essential to produce the phenomenon which we call flame, or the combustion of fuel. When all the elementary substances are present, two things appear to be necessary to effect combustion, and these are heat and the requisite quantity of oxygen as its supporter. Now this latter element is supplied in great abundance from the atmosphere; but as it is mixed with another gas, nitrogen, from which it has to be separated before it unites with the hydrogen and carbon of the fuel, it becomes absolutely necessary to maintain, not only a high temperature in the furnace, but to afford facilities for the requisite supply of air under all the varied conditions of slow to active combustion. As the evolving gases will combine with no more than their correct equivalents of oxygen, it becomes a question of great importance to approximate as nearly to the right quantity as possible, and to admit neither more nor less air than is necessary to furnish those equivalents. If more air is admitted than what is required, the temperature is reduced; and on the other hand, if too little air is admitted, an imperfect combustion ensues, and the usual defect of a turbid black smoke is the result. When the proper quantity of atmospheric air is supplied, and a sufficiently high temperature of the furnace is maintained, a perfect process of combustion is then effected: the carbon of the coal, under these circumstances, is converted into carbonic acid, whilst that of the hydrogen is converted into water in the shape of vapour.

"In this state of the furnace the products of combustion become invisible, and hence we may reasonably conclude that smoke, whenever

or wherever it is presented, is neither more nor less than the result of imperfect combustion. A high temperature being therefore one of the conditions necessary to effect combustion, it is of great importance that we should know at all times the varied forms and temperatures at which the gases pass from the furnace into the flues and the chimney.

"Having determined the conditions and relative proportions of the gases and their supporters in a state of perfect combustion, it will be seen that in order to ensure economy and effect in the combustion of fuel, a large and copious supply of air must be admitted to the furnace, and that in the ratio of 15 volumes of air to 1 of coal-gas. It is difficult to determine the exact quantities evolved from every description of fuel, and probably equally so to supply its equivalent of air; but in order to attain certainty in this respect, let the openings be made sufficiently large, and, by a little attention to the quality of the fuel and quantity of air required for its combustion, the apertures may be contracted till such time as a mean average and a close approximation to the maximum effect are obtained."

It will thus be understood from the above quotation that practice and correct arrangement —as before alluded to—are the best teachers.

The proportion of the temperature of the crown plating to that of the sides above the fire bars is not yet decided. The flame—as before stated—beats against the top part of the fire box, but the sides receive a sliding action: this difference of action is simply due to the natural tendency of gases to ascend. The numeral ratio may be taken—approximately—as 1 is to ·5, *i.e.*, the area of the crown surface has twice the evaporative powers of the sides. It must not be forgotten, also, that the height of the space above the fire bars greatly affects the proportion alluded to. The fire box also is not *always filled* with the most effective produce of combustion, the presence of clinkers, imperfect supply of fuel, bad coal, and inattention, all lower the evaporation attainable.

The flame must now be followed—metaphorically—into the combustion chamber. The evaporative powers of the sides, ends, top, and bottom, also demand attention. Now the flame, after leaving the fire box—propelled by the draught—is presumed to fill the chamber in question. There seems to be a doubt, however, as to the reality of this fact.

It can be noticed in the illustration alluded to that the back portion of the shell and combustion chamber is at an angle with the base line. The cause for this is a two-fold purpose: in the one case, the rise of the floor of the hull requires prior attention as to the form, while, in the other instance, the bottom of the combustion chamber is feeble, as an evaporative agent. It will be seen, also, that the water space between the chamber and the shell, at the back of the combustion chamber, is narrow, or from four to six inches between plates. The centre of the radius of the curve—connecting the angular and vertical portions—is at or near the inner extremity of the fire box, or at the bottom of the back tube plate.

The theoretical width of the crown of the combustion chamber is subservient to the cubic contents of the same, also to the beam of the vessel for which the boiler is designed. The actual minimum width at the top may be taken as one foot six inches, increasing to a maximum

of two feet six inches. The former distance is the least available for repairing facilities, while the latter will be ample. The length is due to the number of tubes and width of the fire boxes. Some authorities prefer separate chambers to each set of tubes, and thus decrease the total contents of the former. This latter arrangement increases the heating surface, having extra water spaces introduced between both duplicate compartments.

The cubic contents of the chamber in question must be duly considered before deducing the proportionate surfaces for evaporation. The chamber, it must be remembered, is situated between *two* opposite functions—the one, the fire grate, as the supply; and the other, the tubes, as the discharge. It is obvious, therefore, that the chamber should be of duly recognized proportion to each function alluded to.

To better enable the truth of these remarks to be fully appreciated, the following table of the quantities of several proportions of high boilers are introduced: it may be added, that these examples are in actual practice, and of late construction:—

From this table a concise conclusion can be arrived at without fear of doubt or delusion.

The portion of the present subject next demanding attention is the action of the flame in the combustion chamber. The flame when entering the chamber under notice, inclines towards the tube plate rather than the side opposite. This is due to the draught, therefore the crown or top of the chamber is mostly affected, and the best evaporative portion. The flame has to be contracted or altered in form before entering the tubes, hence a cession of passages must naturally ensue. This fact, to a certain extent, is the basis for the theory briefly commented on in this work,—*see* pages 26, 27, and 28, where it is stated that gases *ascend* if space be allotted, rather than proceed at right angles without rising. It is also very evident that the tendency of the flame is to escape through any opening possible, and that the contraction of the area of the tube opening accelerates the heating properties of the tube or face plate to a certain extent.

Now, with reference to the action of the

No.	Nominal Horse Power.	Total Area of Grate Surface in square feet.	Cubic contents of fire boxes above grates in feet.	Cubic contents of Combustion Chambers in feet.	Total Area of Tubular Passage in square feet.
1*	800	563·76	1008	576	84·5
2	600	414·6	768	544	66·29
3	500	341·5	620	460	51·89
4	400	260·64	499·2	429	41·16
5	350	232·96	432	377·8	33·36
6	200	142·4	253·56	203·62	20·15
7*	150	105·0	186	140	14·19
8	125	79·24	149·4	102·37	12·45
9	100	79·8	134·16	120	10·4
10*	80	57·84	99·84	77·12	12·25

* These boilers have separate combustion chambers to each fire grate.

flame on the tube plate, the upper portion is undoubtedly mostly operated on by the gases of the highest temperature, and the bottom the lowest. Practical evidence of this is seen with the lower rows of the tubes of marine boilers, where they are almost choked, with soot, in a watch of from four to six hours. This fact will not be wondered at, when it is remembered that the smoke—in the fire box—passes direct, under the tube plate connection, to the nearest exit; and the flame being of lesser density, ascends to the central and upper tubes. A further evidence of the disposition of the flame is known by the errors in stoking. When firing at or over the bridge, the *flame* proceeds up the funnel, the evaporation within the boiler being greatly lowered thereby; whereas, by correctly distributing the fuel, the upper tubes receive the purest flame, and a greater expenditure of the heat, before it reaches the uptake.

To render the tube plate the *principal* evaporative agent, the remainder of the boiler must be sadly out of proportion, to say nothing of the waste or consumption of fuel thereby.

Next, assume that the total area of the tube openings or passages is *reduced* disproportionately to that of the grate surfaces, the combustion chamber will be filled with the flame and gases to a *choking* extent, and thus *check* combustion. The natural action of caloric being to ascend, the crown or top of the chamber will be mostly operated on by the flame, and the tube plate next in ratio.

Again, for comparison, presume the total area of the tube openings to be *increased* in proportion to the remainder of the surfaces, the area of the spaces between the tubes will be decreased in due ratio, and therefore affect the evaporative powers of the same. The draught or velocity of the flame through the tubes will be increased also, while the surface of the crown will be less operated on, but with an increased temperature due to that of the flame.

Now, to still further exemplify from this natural effect, imagine that the area of the tube openings is *greatly* increased in proportion to the last example—sufficiently to receive, without the least retardation, the produce of combustion from the grate. The result will be that the surfaces of the tube plate, crown, back, and ends of the chamber, are greatly lowered in temperature. It may now be presumed that as the tubes contain the full effect of combustion, they will naturally give out more heat proportionately. Such, however, will not be the case, due to the simple fact that retention of the flame is required, and that the flame in the tubes generally "slips" along rather than waiting for its heat to be extracted.

It next is requisite to consider the relative proportion of the different parts of the combustion chamber.

It will be remembered that allusion has been made to the requisition of *air* to produce combustion; this can be further verified by an act of universal domestic practice, viz., "the lighting of gas or a candle;" here we have a vivid truth. The gas and the oil are not igniferous *alone*, but the amalgamation with the atmosphere is imperative to produce ignition, and indeed it forms a large compound, and simple as this is, it will be difficult to produce a more striking manifestation of the truth of natural laws.

resuming that the *correct* amount of air is g admitted, and the proportions of the openings and capacity of the chamber to irea of the grate surface are of *proper* ratio, evaporative powers of each portion of the oustion chamber, numerically considered, be known by—

:own surface	1
ibe plate surface (effective) ...	·875
ick (from crown to bridge) ...	·6
ids (from crown to bridge) ...	·5

ill thus be noticed that the surfaces below grate—from the front of the fire box to bridge—are considered non-effective for oration.

ie flame, or rather the effect of the same, ig thus far been followed, the tubular ce as an evaporative agent must next be dered.

ie portion now under notice is cylindrical orm, and therefore the *best* adapted for gth. This being a fact patent to all engi-, boiler tubes are rarely made more than ighth of an inch in thickness, whereas the inder of the boiler is from *three*-eighths to -eighths of an inch, to say nothing of the isition of the stays. Of course the diameter the surface of the tube exposed to the ure is duly considered in relation to the : portions, in this instance of comparison. ust be remembered that the entire proceed, the grate, passes through the tubes, hence esult will be in due proportion thereto. tube can only receive a given portion of and gases, also its surface is *alone* affected ie pressure surrounding it. The remain-portions of the boiler are *connected*, and by a simultaneous resistance is imperative.

The action of the flame and heated gases, in the tube, is not equal on the total surface, due of course to natural laws.

A practical evidence of this can be seen by holding a lighted candle at the extremity of a tube, the flame has a great tendency to strike the upper portion of the circle, in a line for a given length, before conforming to the circular shape. Now, the flame from the combustion chamber acts precisely the same in principle, with the additional effect due to the relative quantity. The surface mostly acted on throughout the tube's length—according to natural laws—is the upper half. As however the tube is inclosed, and the exact velocity of the flame cannot be indicated, or even seen, another fact presents itself for due notice: "The speed of the flame is due to the draught or quantity and pressure of the air admitted in the fire box." It is therefore certain that some portion of the air enters the tube with the flame and remainder of the gases. Now, the effect of the flame on the inner surface of the tube is in direct proportion to the velocity and nature of the caloric. Presume an ordinary rate of draught to be admitted, the surface, exposed to the flame, will doubtless be thus operated on. The first half of the tube's length may be considered to receive the most effect on the upper half, the next portion, taken at one third of the total, —by the draught forming the flame into a tube, or hollow,—receives the effect of the flame on the entire surface. The remainder of the tube's length—as the draught is lessened —may not be fully exposed to the flame throughout: and those theories are based on the following laws.

All gases, when accelerated in their progress by draught, conform to the upper, rather than to the lower, surface of the opening passed through, due of course to the *air* passing *under*, rather than over, the gases.

All fluids and gases—due to natural laws—when exposed to an equal surrounding pressure, assume a tubular form, rather than any other. For example, the termination of a fall of water or ascendency of a jet, also the flame of a candle.

The flow of gases, from contracted to expanded receptacles, always ascends as it proceeds, due of course to the air passing under the gas: the flow of fluids is the reverse in action, but alike in principle, gravitation being the cause thereof. Now these facts, being obvious to all, the conclusions given, as to the action of the flame in the tubes, can be readily understood.

The next consideration is the value of the tube surface operated on and the different temperatures throughout the same. The end of the tube nearest the combustion chamber is the first part operated on by the flame, therefore the temperature of that portion may be said to exceed that of the remaining. It must be remembered, however, that—according to the theory of the action of the flame—the surface, in actual contact with the flame of the greatest heat, is the upper half for a given length, therefore is an accelerator or transmitter of caloric to the remaining portions. It is obvious, then, that what is lost by contact is rendered by temperature, and what is obviated by a lesser heat is gained by surface. To better appreciate these facts is to consider the conducting powers of the material operated on, and the rapidity of the "traverse of the heat," in proportion to the substance of the tube.

The materials used for the construction of marine boilers are copper, brass, wrought and cast iron, lead, brick, wood, and felt, the two latter in this case for the retention, rather than the production, of heat. Now in all branches of science the consultation of natural laws will, when understood, ensure a correct result. It is obvious therefore that, to justly investigate the present subject, each material adapted for evaporation must be separately noticed, particularly as to its powers of conduction. According to credited authorities, the following may be considered as the most accurate proportion yet attained.

Name.	Conducting power for transmission of heat.
Copper	1,000
Brass	468
Wrought Iron ...	336
Cast Iron	311
Lead	161
Brick	9·88

It will be seen that copper and brass are the two best materials for evaporation, hence their introduction for tubes. The wrought iron, forms the greatest portion for the construction, but in some cases the shell of the boiler has been made of copper, while in others the fire boxes only have been of that material. The cast iron forms the fire bars, external fittings, &c., the lead being mixed with other metals for fusible plugs or jointing. Brick is used chiefly for the bridge, but in some cases it has been introduced to line or protect some portions of the fire boxes. Wood and felt constitute the lagging, or the external covering for the sides, back, and top of the boiler.

return to the effect of the flame in the tube, a simple but important fact now presents itself for due notice, *i.e.*, the conductive nature of the material and the thickness or thinness of the same. Another natural cause is also worthy of consideration. "The rapidity of the penetrative traverse of heat on any material is in due proportion to its *thickness;*" hence in the case of tubes, the conducting powers being at right angles in action, the portion receiving the greatest heat, actually, affects the surrounding portion to a great extent, and thus an entire *evaporation* is maintained throughout. From these results it may be justly stated that the *entire surface of the tube is an evaporative agent* —not as the fire box, or the combustion chamber, where some portions are *non-effective.*

Having thus far digested the present subject, attention will now be given to the values of the heating surfaces of the various available portions alluded to. From the table given of the conducting powers of the different materials, it is seen that the value of copper as an evaporative agent, is 2·976 to 1 of wrought iron.

Now, assuming that the caloric is of a lower temperature within the tubes than in the fire box or the combustion chambers,—which is giving the benefit of the greatest heat to the two latter —the *actual* effect of the flame,—independently of the conducting powers of the metals,—may be said to be greatest in the tubes, due of course to the law of the "penetration of heat." Another fact must also be noticed before deducing the final result; the thickness of the material forming the fire box and combustion chamber, in proportion to that of the tube, which may be considered on an average of 3·5 to 1, and the ratio of the penetration of the heat in each case can therefore be readily understood. It must be remembered, also, that the surfaces of the fire box and chamber are interspersed with laps of plates and angle iron, thus causing a double thermal resistance to that of the remainder. Now, the tubes are of a plain or even surface, and throughout their length of parallel thickness, therefore obviously offer *less resistance* to the penetration of the heat than any other portion available.

The actual proportionate values of the surfaces now under notice, considering the conducting powers, radiation of the metals, and the action of the flame, are as follows:

FIRE BOX.

				Value.
Crown	...	Total surface	...	1·000
Sides	...	Above grate	...	·500

COMBUSTION CHAMBER.

Crown	...	Total surface	...	1·000
Tube plate		Effective surface	...	·875
Back...		Above the level of the bridge,		·603
Ends...		" "	...	·500

TUBULAR SURFACE.

Tubes	...	Total surface	...	2·976

The value of the tube surface is therefore considered to be 2·976 to 1·000 of the crown of the fire boxes and combustion chamber: the decimal equivalents refer only to their respective whole numbers. These proportions will be varied, with given arrangements, to a certain extent, but not to affect the evaporation seriously.

The following table of the areas of the respective surfaces exposed to the flame, will greatly assist future practice.

Nominal Horse Power.	FIRE BOX. Surfaces in square feet.		COMBUSTION CHAMBER. Surfaces in square feet.				TUBES.
	Crown.	Sides above grate.	Crown.	Tube Plate effective.	Back.	Ends.	Total Area in square feet.
800*	664	499·2	115·2	251·5	378	240	13515
600	466·2	350	108·8	105·71	326·4	61·44	10406
500	450	390	92·8	151·11	232	54·4	8560
400	306·24	240	86·4	147·4	216	48	6418
350	268·8	240	76·16	152·28	238	64	4834
200	166·4	133	41	72·15	117·9	27·6	3040
150*	120·71	93·1	19·95	43·26	67·5	64·5	1886
125	106·56	80	22·12	31·93	53·32	14	1671
100	87	71·8	25·73	43·85	62	14	1116
80*	84	70	15	24·1	54	24	1440

* These boilers have separate combustion chambers to each fire grate.

The general practice—when these proportions are known—is to consider the surfaces of the grates and tubes only. This of course will only be practically correct when the relative proportions throughout are carefully produced by the given arrangement of the high boiler.

The situation of the flame, when last alluded to, was at the front tube plate; there its evaporative powers—theoretically—are ended, but it is proved in practice—and that too often—that such is not the case. Indeed, proofs of this denial may be often seen from the cause of ventilating casings surrounding the uptake and funnel.

The longitudinal form of the smoke box and uptake within the boiler, represented by the illustration, Fig. 12,—page 83,—is not a universal arrangement. It must be added that the position of the super-heater, in this case, is the main consideration. In the event of the super-heater being independent of the uptake, the latter is situated at a greater distance from the front plate. In that instance, the steam surrounds the uptake, and it is actually a super-heater in principle. Water spaces are also sometimes formed between the smoke box doors, in order to absorb the heat during its final exit.

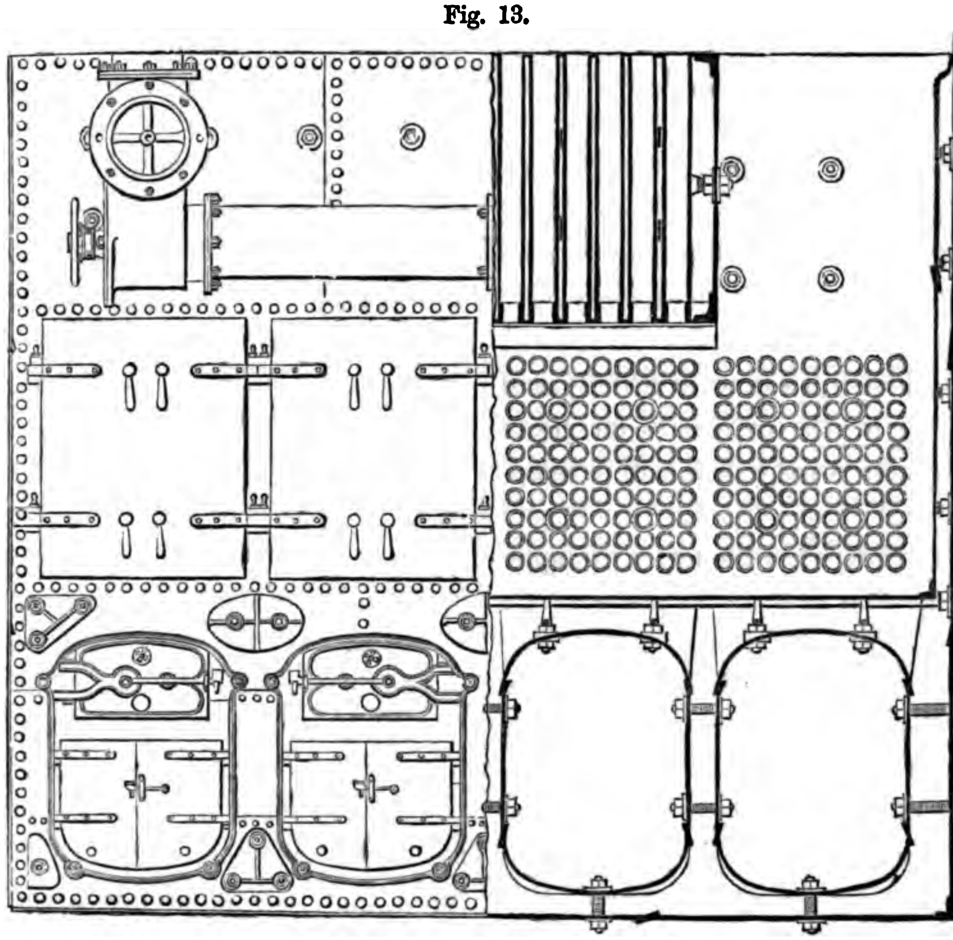
It will be noticed, also, that in the arrangement represented by Fig. 13—page 93—the uptake extends above the boiler, for its length. In other examples, the portion, for the final exit of the smoke is, only, above the roof or top of the boiler. The uptake in these cases commences slightly above the tubes, and increases in height—angularly—in proportion to the area of the passage required, the final portion being perpendicular, and the super-heater beyond the same. With a third arrangement, the height of the boiler is increased to receive the super-heater within the smoke box, the latter being below the top of the boiler; such a disposal is termed "extra high"—the raised portion extending to the deck of the vessel, and, in some instances, above the same.

The actual temperature of the smoke box depends entirely on its proportions. With a narrow bottom space, below the tubes, and perpendicular front, the doors are made with air spaces to counteract the effect of the flame, and in

ə case of a proper width, as in Fig. 12—ge 83,—the doors can be arranged as there thout fear of destruction. The progress of ; flame as it leaves the tubes is, of course, as-ıding, forming a curve. Hence, with cramped ɔke boxes, the effect is to retard the flame, l thus extra air spaces are introduced.

purpose of evaporation. It may now be argued that the same action in principle occurs in the tubes—which are in the present case considered the main agents—and thus a similar effect will be the result therefrom. The simple but truthful law must again be alluded to: which is that all gases naturally tend to ascend, but, in

Fig. 13.



ELEVATION OF THE BOILER SHOWN ON PAGE 83.

Another fact also presents itself for comment: ; flame when thus retarded—in the smoke ɛ—during its traverse, checks combustion the fire box, thereby producing a total loss əffect.

The action of the flame in the smoke box is a sliding nature—of the least effect for the

the case of the tubes, a horizontal traverse is maintained, being at right angles to that in the smoke box. The ratios of the heating properties of the tubes and the *sides* of the smoke box, with a correct proportion, may be considered as: value of tubes 1, smoke box ·125 to one-tenth and ·0625, in extreme cases. It must be

noticed that, with narrow smoke boxes, the effect of the heat is retained within the same more in proportion than those of a wider construction. It now becomes requisite to decide which is the better mode—relative to the requisition of air, casings, and doors. The matter resolves itself into simple conclusions, *i.e.*, to check combustion, is to reduce the effect in the fire box, and thus lessen the value of the heating surfaces throughout. With an ample smoke box, it is therefore certain that the heat is more fully expended before arriving at the same, than with a confined arrangement, and thus the temperature of the final expenditure lessened in proportion. To be concise as to the observance of natural laws in relation to the marine boiler, it can be truthfully said that no portion of the desiderata for steam navigation is more worthy of future development.

Now, with reference to the many examples of the arrangement of high boilers, they are generally, in design and arrangement, as that represented by Figs. 12 and 13—pages 83 and 93. The boiler with the plain and flat top is chiefly used for war vessels, the "extra high" being for the mercantile navy. It is preferred by some makers to dispense with the water space directly under the fire grate, and thus reduce the total height of the boiler. With such an arrangement the portions of the boiler between the fire boxes are enlarged at the base line, to retain the sediment and permit access for the removal of the same. It may be added that, in either case of construction, the "mud hole doors" are between the furnaces, at the base. The universal shape of the top or crown of the fire boxes are as those illustrated, but in some instances a semicircular form is preferred, which, doubtless, is the stronger.

The form of the combustion chamber is much the same in all instances, except in the case of the back of the boilers being perpendicular from the base line, when the chamber is arranged accordingly.

The length of the tubes, in all instances, depends on the beam of the hull, when the boilers are transversely arranged in the same: while in the event of a longitudinal position, the tubes can be of maximum length, space being duly considered.

The smoke box having had prior attention, it will be sufficient to state that, for the national navy, the low smoke box is imperative, due to the rule: the roof of the boiler must be, at least, one foot below the line of flotation. In the mercantile navy the extra high boiler is mostly adopted; with this class, too, the smoke box and uptake can be vertically elongated—and the shell of the boiler likewise, to surround the same.

The form of the shell is, in many cases, as that illustrated. In some instances it is preferred to curve the sides where connecting with the top, while, in other examples, elliptical and semi-circular roofs have been adopted: these types apply chiefly to the adoption for war vessels. For commercial steam ships, the top of the shell is raised—for a given width—beyond the sides, and thus an extra steam space is produced.

The remaining portions of the boiler now under notice, such as the stays, damper, safety valves, brine trough, and the external fittings, will be fully treated under the heading, "Marine Boiler Details."

THE "EFFECT OF HEAT" IN LOW BOILERS.

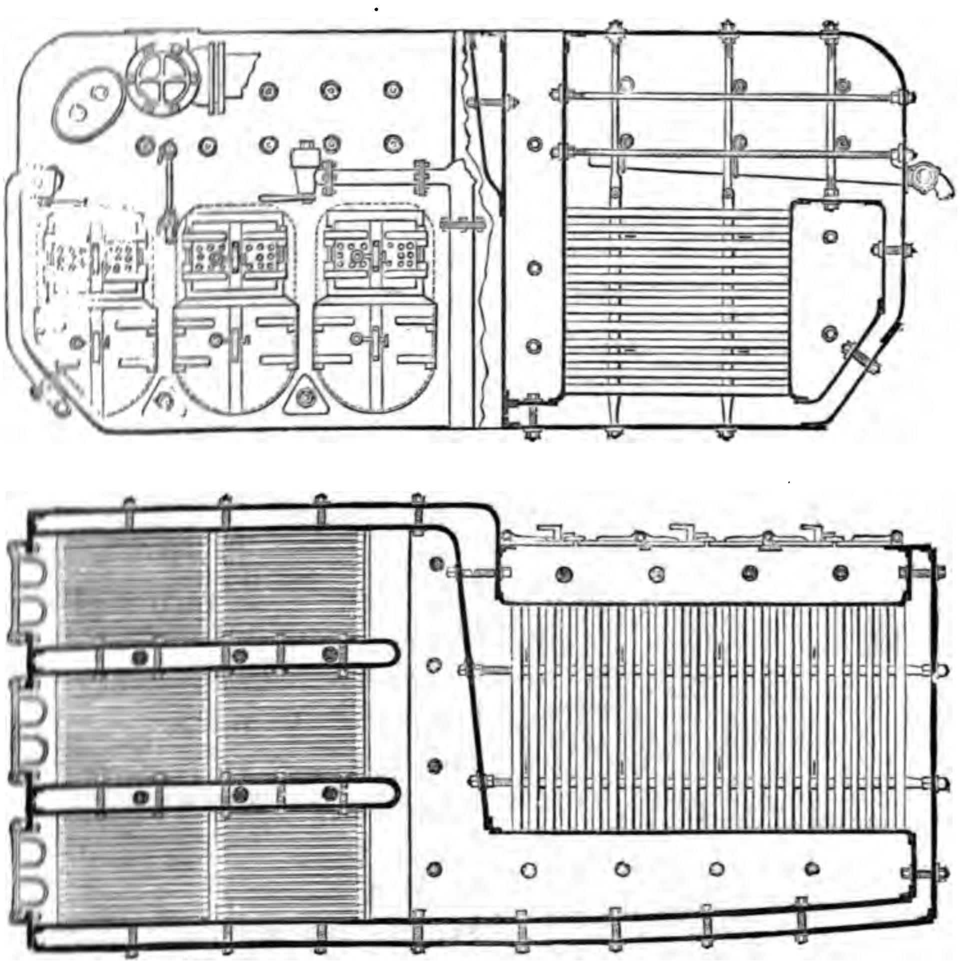
The "law of heat" and the relative values of the heating surfaces of the "high boilers" having been duly commented on, attention will now be given to the requisitions of the low type. The class of boilers now under notice demands more attention as to the internal arrangement than the "high" kind, due to the fact that the allotted vertical space is reduced almost one-half to that already alluded to. This will be fully understood by noticing that, in the case of high boilers, the vessels in which they are fixed are of large tonnage, whereas with the low type, small steamers only receive them. In each case, it must be remembered, the top or roof of the boiler must be below the line of flotation at least one foot. Another feature in the case is worthy of attention—the tubes in the high boiler are situated *over* the fire box; but with the low kind, vertical space being contracted, this arrangement cannot be. The engineer has thus to choose, or rather obey, the cause, to attain the effect. The result of this has been that, to the present, nearly all low boilers have their tubes *below* the level of the crown of the fire box.

To render the present and future remarks on this subject of practical utility, the boilers, illustrated and described, will be those by the principal firms of England and Scotland. Each arrangement and the action of the flame will be consecutively alluded to, and thus ensure a just comparison.

The illustration, Fig. 14—page 96—represents a sectional plan, sectional elevation, and an elevation complete. The type of low boiler now adverted to is the "right angle tube" kind. This term is simply conventional, applying direct to the position of the details. The fire boxes, it will be seen, are at one extremity of the shell—three in this case are introduced. This number is not imperative, as in some instances two fire boxes have been preferred, while in other cases four have not been deemed too many. To be concise as to the correct numbers, the area of the grate surface required and the beam of the hull are the precedent considerations. The combustion chamber in this instance is transversely and longitudinally arranged, relative to the fire boxes. It will be noticed also that the portion of the chamber, in a line, or a continuation, with the outside fire box, is contracted at the further extremity. Another point, not to be unnoticed, must now be attended to. The area of the fire box, in a line with the combustion chamber, is the least, proportionately with the remainder. The arrangement of the tubes is on a level with the fire boxes at right angles with the combustion chamber. The smoke box is rather confined in this arrangement, due to the beam of the vessel. It may be added that space in this instance is imperatively allotted between the tubes of each boiler for the removal of the same. Space is also required for the opening and closing of the doors when hung on hinges, also the same requisition when removable. There is no great objection to the form of smoke box shown, with the exception that it might be wider, if practicable. To attain this, in some instances the flames from both boilers are received from the tubes into one compartment, the doors for internal access being at the end of the same. The top of the smoke box is curved or bridged from the commence-

ment to the uptake, where steam spaces from each boiler surround the same. With a third arrangement, the bridged top to the smoke box is continued throughout its length, and the uptakes are separate *within* each boiler. A judicious arrangement has been produced, by making the shell of the boiler nearest the side of the hull straight; this, of course, when space is available. The tube plates are angularly secured, and by that means, contract the combustion chamber at the uptake extremity, and enlarge the smoke box at the grate end.

Fig. 14.



GENERAL LOW BOILERS, AS FITTED IN THE ROYAL NAVY.

Now, as to the effect of the flame in the boiler now illustrated. To know that, a practical conclusion, as to the operations of combustion and evaporation, must lend its aid: which conclusion must be the result of great experience. Presume each of the grates shown in the plan of the boiler above illustrated, has a similar quantity of air, what will be the result? The two furnaces nearest the centre line of the hull, also a portion of that at the side of the same, are faced at right angles with a wall of impenetrable structure,

as far as the flame, gases, and smoke are concerned. The flame, on reaching this portion—which is the divisional plate between the combustion chamber and the tubes—beats against it in a direct line. It is obvious, therefore, that a sudden shock to the current of the flame's progress must ensue. In fact, it is doubtless the case also, that there is a recoil from the shock alluded to: but the effect of the combustion, from the grate at the side of the shell, is more directly conveyed into the chamber fronting the tubes.

It is next requisite to consider the *progress* of the flame and gases when meeting at the commencement of the chamber nearest the tubes. Now, it is well known that the velocity of the flame from the grate to the combustion chamber, and thence through the tubes, is in due proportion to the draught attainable. It is obvious also, that gases proceeding in a direct line are exposed to less friction than those proceeding through angular and curved passages. Again, for comparison, gases operating on a surface, at right angles to the line of progression, are greatly checked in velocity. From these natural laws the matter under notice resolves itself into a simple fact of conclusive evidence. The flame raised in the grate, at the side, passes direct to the outer extremity of the combustion chamber. The flames from the next grate strike mostly on the corner of the tube plate connection, and also displace a certain amount of flame from the preceding furnace. The grate nearest the centre of the hull requires more air—as draught—than the two former. The flame at the outset, in this instance, is formed into two currents; the one very fast and the other slow. This will be duly appreciated on remembering the natural flowing of fluids and gases. The progressive current passes at right angles to the grate until reaching the next current, when both amalgamate to a certain extent. The line of actual division is *not* entirely lost, due of course to the fact that, the temperatures of the flames emitted from the respective furnaces are unequal, also in density and velocity. The passive portion of the flame from this furnace, nearest the hull's centre, is that which is less rarefied than the remainder, consequently it is lodged in the angular portion in front of the respective grate. In order to eradicate this inequality or rest of motion, the connection of the side of the grate with the divisional plate has been curved, and thus the evil alluded to greatly mitigated.

The *action* of the flame on entering the portion of the combustion chamber in front of the tubes, next demands observation. It must now be remembered that three distinct volumes emanate from their respective sources, and are received into a single compartment. The congregation of the gases is most at the grate end of the chamber in question, therefore the tubes at that locality will receive the greatest volume. This ratio of reception, within or through the tubes, is due to another cause in operation in the smoke box; which is, that as this volume is situated furthest from the uptake, it is obvious, then, that the remainder of the flame traverses in front of the remainder of the tubular passages, and thereby tends to check the progress of the flame behind, and from that fact it can

be readily understood that the tubes nearest the grates are mostly operated on by the flames, and the remainder in consecutive proportion. The extreme end of the combustion chamber receives the flame emanating from the fire box in a line with the same, as before alluded to. The middle fire box contributes to the portion or set of tubes next in position, and the remainder is filled from the proceed of the last or inside fire grate.

It may now be presumed, contrary to the views now quoted, that the law of combustion being that "draught propagates heat in the proportion that caloric produces evaporation." The flame from the side furnace is the *first* to enter the tubes. Now, to attain this, the flames from the remaining grates must *cross* each other, and to further test the truth of this, the laws of forces must be considered.

It is well known that the velocity of flame is in proportion to its rarefication and heat; also, that the greater the draught, the greater the resistance required to impede the flame duly affected. Thus, if a volume of gas of a given density is proceeding in a direct line at the side of a plate or structure, a second volume from an independent source—of lesser density—striking the first bulk direct or angularly, displaces only a slight portion, if any, of the same, and the progress is not materially affected. The second volume proceeds towards the line of exit, rather than amalgamates with a bulk of gas of a lesser rarefication. These laws may be presumed to convey the idea that the gases do not mingle within the combustion chamber; but it will be well to state, that the amalgamation is due to the capacity for that process, also the draught and area of the tubular passages.

Those preceding remarks strictly apply to the action of the flame within the combustion chamber of the type of boiler now under notice. The flames from the two inside grates, or nearest the centre of the hull, have to pass through a given space. There is also a certain resistance or frictional contact to be overcome. The effect of this is "time," for combustion and rarefication before reaching the tubes. The flame from the outside fire grate passes *direct* from the same to the front of the tube plate in a volume of greater density than that first alluded to. There being no resistance in the present case, the velocity of the flame will be the greatest also for a given period.

The remarks to the present apply directly to the "action of the flame," with a certain arrangement; but the vital consideration is the "effect of the action," to which attention will now be given. It will be superfluous to recapitulate the conducting powers of the materials. It is sufficient, therefore, for the present purpose to state that the preceding remarks in allusion to the nature and thickness of copper and iron apply to all examples of marine boilers.

The arrangement of the boiler in plan, Fig. 14,—page 96—must now be remembered, to assist conception. The flame in the fire box nearest the centre of the level, or inside, strikes the crown of the furnace in a series of curves for a given length, direct to the bridge. The crown of the combustion chamber is next affected, and, thirdly, the front or divisional plate *above* the bridge. The flame from the

next grate operates on the crown surfaces similar to the first, but the front plate being exposed to a greater volume of gases is thereby doubtless evaporatively effectual *below* the bridge of the second grate. The flame from the outside grate may be said to bound on, or strike, the furnace and chamber crowns more than other portions, the latter being of course the least effective at the back end.

The sides of the second furnace are the most effective, in relation to the others,—the portions above the fire bars, and below the crown, being alluded to. The first half of the tube plate is the most effective, while the remainder is of less value proportionately. The side of the combustion chamber opposite the tube plate is of less ratio than the latter, the end being the least in value of all the portions alluded to. The tubes are the consecutive conductors of the flame to be specified as to their powers of evaporation. The cluster in plan may be divided into three parts—*first*, those nearest the grate—in an angular direction, the greatest horizontal line being the top—are mostly filled with the flame; *second*, a parallel angular portion is next affected, and the remainder mostly on the top portion of the plate. The action, of the flame within the tubes, may be considered as that for high boilers.

The effect of the discharge of the flame into the smoke box must next be alluded to. The volume naturally, in all cases of exit, ascends; hence, in this case, the crown is mostly operated on. The side and tube plate are much the same in effect, the flame sliding from—rather than acting against—these portions. The fire grate end of the smoke box is the least effective of all the surfaces throughout the arrangement; also, the opposite portion at the uptake will not be of any value worthy of recognition.

Thus far the action and effect are obvious, the proportionate values being from these deduced. Before, numerically, giving the result, the equivalent relations of the crown, tubular, and other surfaces must be noticed. The crowns of the fire boxes and the combustion chambers may be considered as the same in value throughout their lengths, the different temperatures in each being duly noticed. The divisional and front tube plates may be taken as the same in value. The end of the combustion chamber and the crown of the smoke box are alike effective. The tubular surface is, of course, of maximum value in proportion to that of the remainder. The following ratios are the actual results taken from practical evidences:—

		Value.
TUBES.		
Total surface		2·976
FIRE BOX.		
Crown ..	Total surface ..	1·000
Sides ..	Above fire bars ..	·600
COMBUSTION CHAMBER. (Portion at front of grates.)		
Crown ..	Total surface ..	1·000
Divisional plate..	Total surface ..	·875
End ..	Above bridge ..	·250
COMBUSTION CHAMBER. (In front of tubes.)		
Crown ..	Total surface ..	1·000
Tube plate..	Effective surface ..	·875
Back ..	Total surface ..	·616
End ..	Total surface ..	·200

SMOKE BOX.

				Value.
Crown	..	Total surface	..	·200
Sides	..	Total surface	..	·125

Those proportions of the value of each portion in contact with the flame, are subject to variation, due to the proportion of capacity of the combustion chamber, and area of the tubular passage to the grate surface. Also, the draught and quality of the fuel cause a varied effect. The proportions given, however, are the *correct* mean results that were taken from actual practice.

It will be noticed in the elevations of the boilers, illustrated on page 96, that the connection of the outer side with the bottom is angular. This form is subservient to the shape of the hull space that receives the boilers; while in other instances these outer sides of the boilers are perpendicular from the base line, and the bottoms are angular. With this form the heights of the two sides are unequal, the greatest being at the centre of the hull, or at the smoke box. The water-space below the tubes is also of an uneven depth, the least space being at the combustion chamber tube plate.

It is next requisite to consider the better form for evaporation, &c. With the section illustrated, the contracted portion is at the base line, which doubtless is the correct place. This of course is obvious when the areas and positions of the tubular passages are considered, also that the natural action of the flame is to fill the upper rather than the lower tubes. It is very evident also that there is actually a gain with the arrangement under notice, *i.e.*, the flame will be compelled, at the contracted part, to act on the tube plate, and opposite it also, and thus increase the evaporative effect. Now, the action of the flame on the tube plate, in the combustion chamber, having a vertical back, is not thus accelerated as with the angular form; but there is one practical fact, however, that should not be overlooked, the combustion chamber with the vertical back, admits of more access for repair, or closing the tubes, than any other form. It will thus be understood that the relative values in the two sections of the chamber now under notice consist of evaporation in the one case, and access for repair in the other.

The connection of the crown with the back is preferred to be curved in some instances, rather than at right angles, as shown by the illustrated section.

The next consideration demanding attention is the proportions of the heating *surfaces* of the type of boilers now under notice. The ratios will not in all examples be alike due to one particular cause, viz., "access for repair." Thus, a boiler with two grates may not be arranged to produce the same proportions as that with three grates. The cubic contents of the combustion chamber, relative to the grate surface, may be increased in the one case, and lessened in the other. The area of the divisional plate to that of the grate may also be reversed in ratio, due of course to the width of the combustion chamber, length of the tubes, and width of the fire-boxes. The effective area of the tube plate, also, may be likewise effected.

Practical statistics being truthful evidences in all cases, whether scientific or commercial, a tabular statement of the proportions of

each arrangement, in actual practice, will be found at the termination of this chapter, and from it a truthful conclusion as to the actual result will be certain.

RETURN TUBULAR—SIDE-FLUE—ARRANGEMENT.

The arrangement of boilers now to be explained—as those before alluded to—is due to the transverse section of the hull for their reception. Now the position of the details in the shell last described and illustrated, was particularly adapted for available space—transversely rather than longitudinally, or fore and aft—of the hull. In cases where the beam of the vessel is reduced in proportion to its length, but the depth or line of flotation from the floor remains the same, the arrangement of boiler, represented by Fig. 15—page 103—is sometimes resorted to. With this example it is seen that the tubes are in a line with the fire box, hence the term "return tubular" is applicable. It is obvious, also, that a much greater *length* of tubular surface is attainable with this arrangement than with that illustrated by Fig. 14 on page 96. Before, however, proceeding further with the present description, it will be better to allude briefly to the correct length for tubes. The subject now under direct allusion is one of the utmost importance: this is obvious when the values of the heating surfaces are considered.

To deduce a correct result from evidence of any kind, an observance of natural laws is imperative. The transverse section of the tubes mostly adopted is the circular kind; but in some instances those of a parallelogrammic shape have been introduced. The gain, by the adoption of the latter, is presumed to be contact of the flame with the surface exposed. This is correct in principle and satisfactory in practice, with the exception of repair and manufacture. The circular tube, particularly when of brass, can be formed, drawn out, in one portion without seam or connection of any kind throughout its length: but the flat or sheet-flue is formed of separate portions connected by riveting, hence the construction is complicated, while the repair is uncertain. Again, for comparison, the circular tube is the strongest of shapes, while the flat kind is the weakest. The thinness of the material in the one case forms a remarkable contrast with the required thickness of the other. The values of the heating surfaces, therefore, will be duly affected, although with the flat tube the flame is more contracted than with the circular form. The action of the flame in each instance is much the same—*i.e.*, the upper portion receives the greatest effect of the flame, hence the better evaporative surface.

Now, with reference to the length of the tubes, so as to produce a correct proportion. A great deal depends on the amount of surface determined on. When this is decided the matter resolves itself into simple calculation: imagine for the purpose of exemplification, a given number of tubes are known in proportion to the area of the passage required, this simple formula will be sufficient; $\frac{S}{C} = L$, when $S =$ total tubular surface, C total circumference of the tubes, and L the length.

Attention must now be given to the natural flow of gases through tubular passages. Next, the velocity of the gases demands notice;

because the action of the former law will be in direct proportion to the latter, and both depend on the density and acceleration attainable. Now, the density of the gases is due to the acceleration,—the latter being in the form of draught, or air, and combustion. It will thus be understood that the final effect is due to many causes, and that each succeeding law depends on the precedent result.

The flame-friction within the tubes also, is in due proportion to the density of the gases. From this fact some authorities advocate the short tubes and large combustion chamber, while others entertain the idea that a long tube will give the better result—the same total amount of evaporative surface in each case being duly recognised.

Now to form a just as well as a practical conclusion: presume two sets of tubes of unequal lengths in separate boilers, but the remaining proportions to be the same, the tubular passages excepted, which will be increased with the minimum length of tubes, also the quantity of air admitted to be the same in each chamber. The combustion chambers will therefore contain an equal amount of flame and gases: and the area of the tubular passages being the least with the longer tubes, the contraction of the flame will be more in that respective chamber than in that of the shorter arrangement.

Now from this cause, the effect will be that, the air admitted has a greater power on the flame in the chamber behind the tubes of the lesser length, and thus the velocity of the flame through them will be the *greater*. Another fact is also worthy of notice—the different flame—frictions that occur in *two* localities at the same time: because it is well known that the alteration of the flame from a common bulk into tubular forms *must* cause a certain amount of friction on the tube plate. It is therefore apparent that the *greater* the number of the openings, or total area of the tubes, the *less* the plate friction will be. It is obvious also, that the less the *retardation* of the flame within the combustion chamber, proportionately will be the velocity of the flame within the *tubes*.

Now the action of the flame with the longer tubular arrangement is in direct opposition to that just described. The combustion chambers, it must be remembered, are the same in cubical contents, but the *area* of the tubular passage is the least in this case; it is apparent, then, that the friction on the tube plate will be the greatest, due, of course, to the contraction of the exit; for the flame and its velocity *within* the tubes is also affected, while from the *retardation* in the combustion chamber *time* is allowed for the *expenditure* of the heat. To be concise, the more time allowed for combustion the more heat is produced.

From these facts *practical* evidence is given that the *long* tube is preferable to the short, although the velocity of the flame with the latter is the greatest, but the *absence* of sufficient time for combustion greatly reduces its evaporative effect.

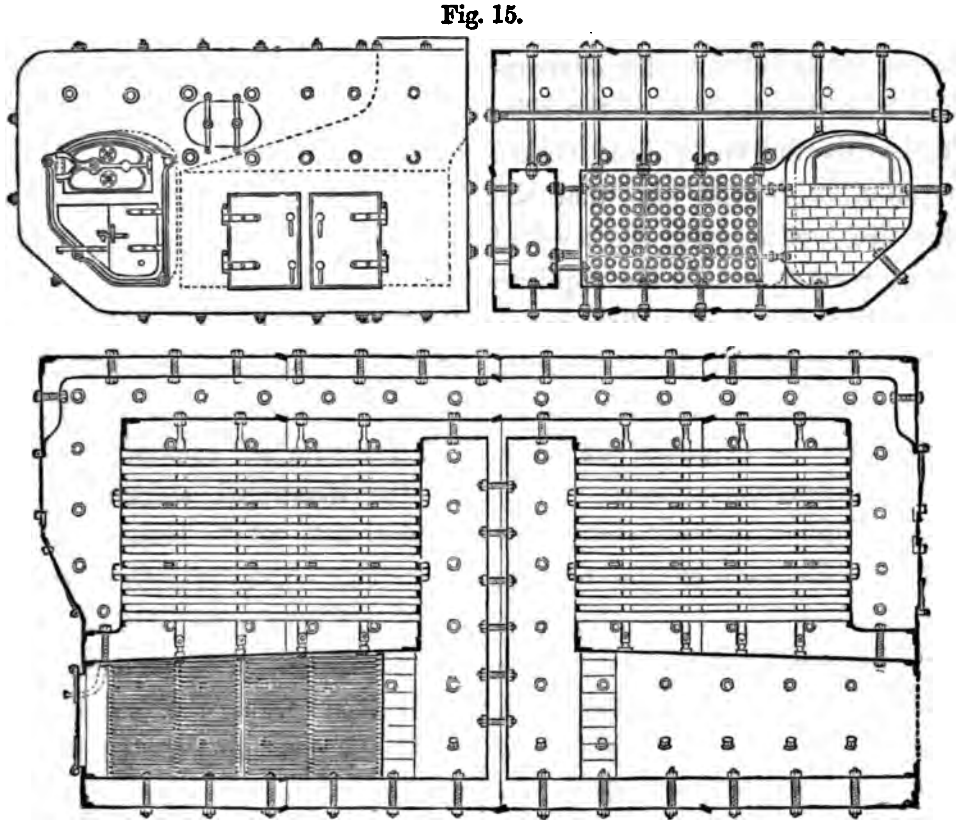
The temperature of the caloric emitting from the long tube is, of course, less than from the short, and from this fact some authorities insist that imperfect combustion is the result. Now it must be understood that smoke can *only* be generated in the fire box, and as the direct communication is to the combustion

chamber, it is in the latter that the combustion —almost if not complete—should be effected. The actual gain, therefore, with correct proportion is, the greatest effect with the most simple cause—*i.e.*, the more perfect the *consumption* of the fuel before the flame enters the tubes, the greater the evaporation proportionately, quantities of coal and water being respectively considered.

shell at the side: and the combustion chambers, one to each grate, are at right angles with the fire boxes; the tubes are arranged, outside each fire box, in a line with the same, thus forming a "return" action. The smoke boxes are virtually separate, at the extremities of the tubes.

In order to render the final uptake common to the smoke boxes, a longitudinal flue is

Fig. 15.



MESSRS. MAUDSLAY'S SIDE-FLUE MARINE LOW BOILER.

introduced, extending throughout the length of the boiler; and on referring to the sectional elevation, it is seen that the flue in question, is the same height as the smoke box, a divisional water space causing a complete separation. The fire box, it is seen, is contracted at the base line, also the side and bottom of the shell are angularly connected. This form of construction is due to the beam

The description of the arrangement now under direct notice must be proceeded with. The above sectional plan and elevations, sectional and complete, represent a low boiler, especially arranged for given purposes, as before explained. The position of the boilers in the hull is represented by the elevations, the plan referring only to one portion, or boiler. The fire boxes are fore and aft of the

of the hull for which the boilers were designed.

It is preferred, in some instances, to reduce the width of the boiler by constructing the longitudinal flue—open at the side—from the crown to the base, throughout the length. By this formation, the central water spaces are dispensed with, and *one* flue is common to the smoke boxes of both boilers, instead of a separate communication, as illustrated. How far this arrangement is judicious for evaporation is obvious from the abolition of the side-heating surface of the flue under direct notice.

The illustration represents also that the crown of the fire box is above the level of the tubes. The cause for this is the requisite height or capacity above the fire grate. Now, when the longitudinal flue is common to both boilers, the grate can be lowered in the fire box, and thus the height of the latter is reduced proportionately.

One of the advantages in the arrangement illustrated, is the situation of the uptake. This is of the greatest importance when two types of boilers, high and low, are used in the same hull. The funnel is situated between the boilers in question, and thus one final uptake is common to both arrangements. It is obvious, therefore, that, what is a gain in the one case is a loss in the other—*i. e.*, the central position of the longitudinal flues has obviated the same position for the fire boxes; therefore first consideration is given to the communication with the aft end smoke box, and the fire boxes are arranged to suit that requirement.

To reverse the positions of the fire boxes generally in marine boilers will be to disturb the position of the chimney; but the present form now under notice admits of a rearrangement internally, without shifting the locality of the uptake.

This will be readily understood, by supposing the shell of a boiler is as that illustrated; and it is required that the fire boxes must be centrally situated in the hull of the vessel: then the only *actual* difference in the "imaginative" and "illustrated" arrangements is, that the fire boxes will occupy the locality of the longitudinal flue, represented in the sectional elevation. The flue in each of the supposed arrangements is at the outer sides of the shell, and the final uptake ascends —*centrally of the hull—over* the fire boxes at the forward end of the boiler. The total width of the shell is, therefore, not increased by this reverse position beyond that illustrated.

To still further exemplify: the smoke boxes in the illustrated section are nearest the vertical side of the boiler, but in the case of the comparative arrangement they are at the angular, or outer, side of the shell. This altered disposition of the detail can be further understood from the sectional plan illustrated, by *presuming* the centre of the hull of the vessel to be *nearest* the fire box side of the shell, or reverse to what is shown.

Now, with reference to the action of the flame on the surfaces, exposed for that purpose, in the arrangement now under direct notice and illustration—the *form* of the line of progression being now especially alluded to because the effect will receive future attention.

The flame, after striking the crown of the fire box, rebounds into the combustion chamber, which, be it remembered, is at right angles to

the grate and the tubes. The *natural* flow of gases during exit, through angular passages, being *curved*, the plate, forming the back of the combustion chamber, receives a sliding action rather than direct; and from that it is obvious that the extremities of that plate are not operated on by the flame as much as the central portion.

The flame, on entering the combustion chamber, naturally inclines towards the tubular passages. The "line of progression," therefore, from the bridge to the tubes, is doubtless curved at the extremities—*i. e.*, the flame forms curves when entering and leaving the chamber.

Now, the form of the curve alluded to is not in all instances alike; this is due to the proportions of the grate surface to the cubic contents of the combustion chamber and the area of the tubular passage. The quantity of air admitted as draught will also affect the line of progression. With a quick draught, ample chamber, and tubular exit, the flame will enter the tubes *nearest* the fire box rather than beyond. It is obvious, therefore, that due consideration to correct proportions is of the utmost importance with the arrangement now under notice.

It is seen, in the plan, that the tubes at the forward end of the boiler discharge the flame directly into the uptake, and that the smoke box at the opposite end forms a communication with the uptake by a longitudinal flue; so that the flame, on entering the smoke box, naturally traverses towards the line of exit; also here the recognition of natural laws will again assist conception in this way.

In the present example a return action is imperative with a right angular means of exit. All fluids and gases, under any circumstance of transit, naturally form curves, and, with this as a cause, the effect is, that the flame from the tubes assumes a semi-circular traverse into the flue.

Practical evidence of this is seen on examining the boilers, as illustrated, when it is seen that the smoke box doors and end nearest the grate are the least affected by the traverse of the flame, whereas the connection of the tube plate and inner side of the flue, shows evidence of the close contact of the flame with that part.

The flame is now imagined to be traversing through the flue, from the tube plate connection to the uptake. The line of progression is doubtless of an undulated character through the flue's length. The side mostly operated on is that in connection with the tube plates; the crown also receives an unequal effect, due to the line of progression.

The discharge from both clusters of tubes are now concluded to be in one smoke box, proceeding therefrom into the final uptake. At this portion of the arrangement the evaporative effect of the flame's action terminates, but it must be remembered that the truth of this fact will depend on the proportions of the boiler and the mode of stoking, and nature of the coal used.

Difficult as it may have often appeared to form a truthful conclusion relating to the actual course traversed by the flame during its progress from the fire box to the uptake, with given arrangements, it is now stated that a careful recognition of natural laws, an assimilation of the same with *practical*

evidence, together with a knowledge of the whole of the requirements are the only means of obtaining a correct result.

The effect of the flame on the various surfaces is the next consideration. The fire boxes receive the first contact mostly on the crowns, the sides being the least operated on. It is almost needless to state that the surfaces *below* the grate are non-affected by the flame above. The action of the flame on the crown being more direct than on the sides, it is obvious the former is the better evaporator, and also absorbs the greatest heat. The sides, it must be remembered, receive a sliding action from the flame, hence the time and friction are proportionately reduced. From practical conclusions the value of the crown surface is known as 1, while the sides on each side above the grate is as ·5 is to 1.

Consecutively to be noticed is the combustion chamber. The action of the flame, on entering this compartment, affects the crown and tube plate surfaces almost equally. The back does not receive an equal action on its entire surface, due to the line of the progression. The ends are next in value, while the bottom is non-effective as an evaporative agent.

The portion of the flame acting on the crown is the most powerful, and although the entire surface is not in actual contact—the corner beyond the bridge being the least affected—the total surface is considered in value as that of the fire box.

The effective surface of the tube plate is the next consideration. The flame on reaching this portion is checked in its progress, having to alter in form, and divide itself into portions equivalent to the arrangement of the tubular passages.

From this cause time is permitted for the flame to *rest* against the surface in contact, and thus enable a more perfect absorption of the caloric. The value of the effective surface is therefore known to be ·875, being the same as for the tube plates of previous examples.

The contact of the flame on the back of the chamber being of a sliding nature, the ratio of the numeral value is duly lowered.

Practical evidence has long ago proved that the *direct* contact of gases, on vertical or horizontal surfaces, is more powerful than when sliding or gliding along the same. This arises from two functions of natural laws being in due operation—viz., "Penetration" and "Absorption" of heat. The first requires time for its production, and the second is actually the result therefrom. It is certain, also, that flames acting on vertical surfaces in curvous forms impart less heat than when acting in an angular or direct line.

On remembering the action of the flame on the back plate of the combustion chamber, it is understood that certain portions of the surface are not operated on equal to the remainder. It is obvious, also, that the natural tendency of the caloric, in this instance, is to recoil from, rather than penetrate the plate in question. Contradictory as this may appear, however, on reflection the truth will be apparent.

The cause for the peculiar progress of the flame alluded to, is the position of the tubular passages; because the tube openings being opposite the back of the chamber, it is obvious that the flame naturally inclines towards those passages for its exit.

The value of the surface under notice will depend on the draught, to a certain extent, in

an inverse ratio—viz., what is a gain with the tube plate and crown will be a loss to the back and ends.

Now as to the actual value with ordinary practice. It must be remembered that the chamber is filled with a bulk of flame and gases, of a higher temperature than that in the fire box, due to increasing combustion. Recognition must also be given to the fact, that the portions below the bridge and at the extremities are the least operated on by the flame. Considering, then, the values of these portions of the least and most effect, and acknowledging the increased temperature, the value of the total surface of the back of the combustion chamber is ·7.

The ends are obviously the least in ratio. The portion at the tube end of the chamber is of the most value, being more affected by the action of the flame during its discharge. The curves assumed by the flame also lower the value of the portions under notice as evaporative agents. The end, at the bridge is only affected above the same, and that opposite is proportionately operated on. Remembering these facts, the value of the ends of the chamber is known as ·25.

The tubes next are presented for estimation, but a concise observation will only be requisite. Suffice it to say that, on remembering the prior observations in allusion to other boilers, their value is represented by 2·976 to 3.

It is doubtless remembered that, with the arrangements under previous consideration, the tube plates at the discharge end of the tubes were deemed almost unworthy of notice, as evaporative agents. Now, with this present arrangement, Fig. 15—page 103—it is seen that the tube plate at the aft end of the boiler *must* be considered as something when deducing the values of the various heating surfaces. It is obvious, also, that the smoke box partakes more of the requirements of a *second* combustion chamber than in former examples.

The direct conclusion now questioned, is the value of the effective surface of the tube plate at the aft end of the boiler.

When previously alluding to the action of the flame, on the front tube plate, it was stated that the line of progression is of a semicircular form. It is from this curvous traverse that the actual value of the effective surface is produced. This fact is obvious on considering the sectional plan illustrated, it is there seen that a return action is imperative. It is understood, also, that the flame, on discharging from the tubes naturally slides along the plate in question; and from this a greater effect is attained at the aft than at the forward smoke box tube plates.

Now it may be argued that the tendency of the flame, during exit, is to escape from or leave the surface alluded to, rather than act against the same. On reflection, however, it will be obvious that the line of progression is the master agent, and to its action will be that of the flame on the tube plate.

The temperature of the caloric is, of course, at the present stage of discharge, lowered in due ratio to the absorption. The value of the aft tube plate is, considering all the previous facts and practical evidence agreeing with the same, known as ·25.

The remaining portions worthy of attention, for the present purpose, are the crown and sides of the longitudinal flue. The conclusion of

the values of those surfaces is a simple matter when the temperature of the flame throughout the flue's length is known. The actual difference in the degrees of heat at the commencement and termination of the flue is in due proportion to the surfaces operated on, and the length of the same. The action of the flame, as before stated, is undulated, and from that cause the effect will be duly reduced. On justly considering the facts now laid bare, and remembering that the temperature of the flame is reduced also, the surfaces in contact are unequally operated on, the result is that the numerical value of the crown is ·2, and the sides ·105.

The following list of the values of the various surfaces of the arrangement now under notice will render the comparative effect of the flame to be readily understood :—

			Value
TUBES.			
Total surface		... ...	2·976
FIRE BOX.			
Crown ...	Total surface	...	1·000
Sides ...	Above grate	...	·500
COMBUSTION CHAMBER.			
Crown ...	Total surface	...	1·000
Tube plate...	Effective surface	...	·875
Back plate	Total surface	...	·700
Ends ...	" "	...	·250
AFT END SMOKE BOX.			
Tube plate...	Effective surface	...	·250
LONGITUDINAL FLUE.			
Crown	Three-fourth's surface	...	·200
Sides ...	Half surface	...	·105

It must be added, in conclusion, that the proportion of the grate surface to that of the remainder, acted on by the flame, greatly affects the ratio of the values here presented. The amount of air or draught admitted over and under the grate also increases or lowers the values. The table at the end of this chapter is that from which those deductions are made, being the result from examples in actual practice, therefore a correct conclusion is ensured.

DUPLICATE RETURN-TUBULAR ARRANGEMENT.

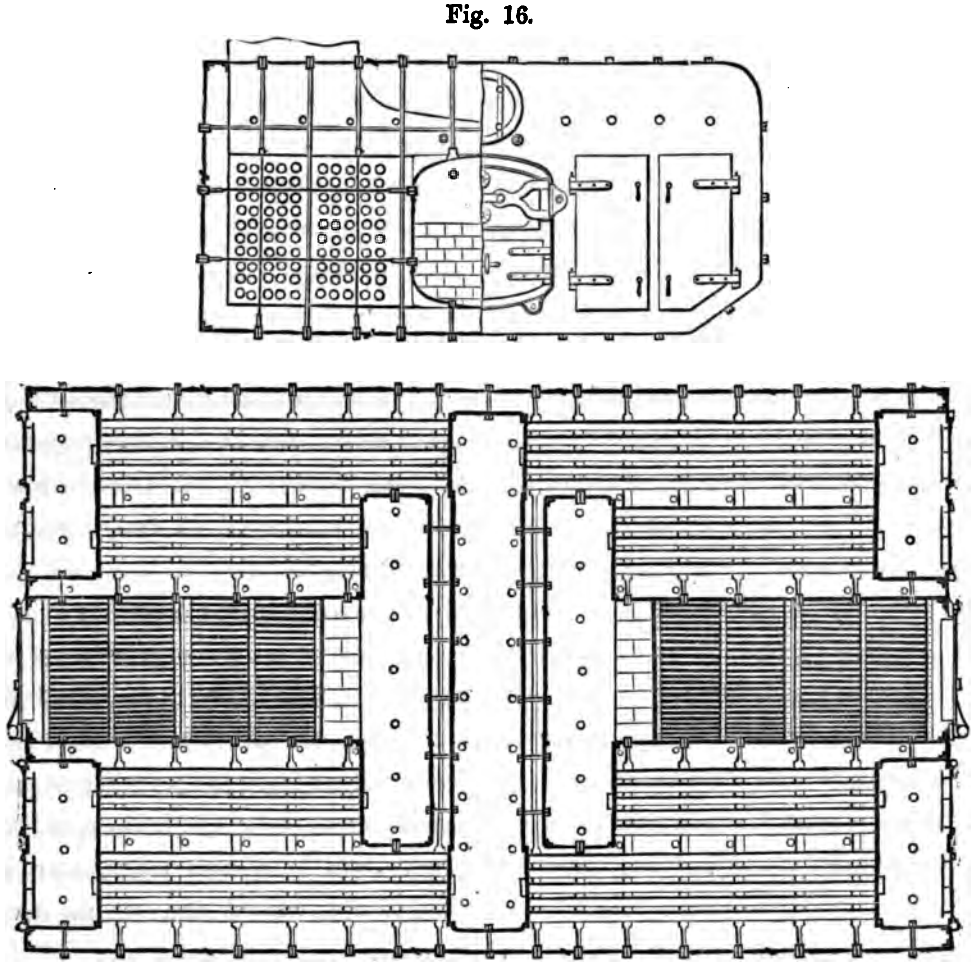
The advantages, as well as the causes, for certain arrangements of marine boilers are only to be appreciated when the requirements are known. The engineer, deputed to produce a given amount of steam power within a certain space, has many matters before him to be duly noticed. The situation of the uptake is doubtless the prior consideration when high and low boilers are required. The matter next in question is the space alloted; not only transversely, or the beam of the vessel, but longitudinally—from the forward bulkhead to the end of the crank shaft. Due attention has also to be given to the amount of coals to be stored, hence the arrangement of the coal bunkers will in some instances affect that of the boiler. The position of the fire boxes is another fact that presents itself for due notice.

When the fire grates are at the *sides* of the hull, the bunkers must be arranged to admit longitudinal room for stoking. It is, therefore, certain that much space is occupied at the greatest disadvantage. Now, when the fire boxes are at or near the centre of the hull, the bunkers can project, from the sides of the hull, much closer to the boiler front, sufficient

space, of course, being alloted to cleanse and renew the tubes. This is obvious on remembering that the length of the tubes is about equal to that of the fire grate; and that the width of the smoke box is added to the longitudinal space required. Also the space required for stoking can be said to be analogous to that for cleansing the tubes, as far as the duties of the operator are concerned.—These remarks apply especially to boilers arranged with fire boxes at each end of the shell, or fore and aft.

It is stated, in allusion to the arrangement prior to this, that, "by a slight alteration internally, the fire boxes could be *centrally* situated,"—the acquirement of this, affecting the position of the longitudinal flue peculiar to the disposition of detail. It will be remembered, also, that the side flue produces a return action of the flame from the aft smoke box. Now, it is in allusion to this latter acquisition that the arrangement at present under notice pertains.

Fig. 16.



BURGH'S DUPLICATE RETURN-TUBE MARINE LOW BOILER.

The above illustrations, fig. 16, represent the plan and elevation of our low boiler, with a duplicate arrangement throughout. The fire boxes fore and aft are centrally placed in the shell. The combustion chamber is common

to separate clusters of tubes, secured on each side, in a line with the fire box. A communication with the smoke boxes, at the extremities of the boiler, is attained by the tubes alluded to. At this portion of the flame's passing in most examples, is the uptake. With the present arrangement, however, it is deemed preferable to produce a *second return tabular action:* the position of the uptake being duly noticed.

It is seen in the plan that, centrally of the boiler is a smoke box or compartment common to four sets of tubes, these latter forming the connection with the end boxes. It is represented also, in the elevation, that the uptake is at the vertical side of the shell. From these facts it is obvious that the final exit of the flame and smoke is at the centre of the boilers longitudinally, and centrally of the hull of the vessel transversely. This position is perfectly correct in theory, and is equally so in practice also, on remembering the natural action of the flame, to which future allusion will be made, when it will be more apparent than now.

Now, it may be argued, in favour of other arrangements having the final discharge of the smoke at the forward end of the shell, that, with the present example under notice, the position of the uptake is a great objection to its general adoption: and further than this, it may be said that when the high and low boilers are in conjunction, the disadvantage is more evident.

The arrangement of the smoke box and uptake represented in plan and elevation, is especially designed for a vessel fitted with low boilers *only;* also, the situation of the funnel and forward bulkhead are imperative. The example was therefore rendered conformable to those prior considerations, hence the objections presumed to be raised are not applicable.

Now, imagine, for the purpose of further testing the practicability of the arrangement under direct notice, that a vessel requires high and low boilers, also that the funnel must be common to both, with the least amount of uptake: and as that is thus far imperative, the remainder has to be rendered complete, with the least disadvantage to combustion and evaporation.

To still further prove that due consideration has been given to all the requirements in each case, conclude that the beam of the vessel is unalterable, or rather, the space transversely alloted for the boilers.

The difficulties to be surmounted may seem formidable, and, indeed, more particularly so on remembering that the smoke and flame must be conveyed to the uptake below the roof of the boiler.

The arrangement necessary to attain the requirements is one of the most simple character. It consists of merely an additional flue communicating from the central smoke box to the uptake at the end of the boilers. This flue can be common to both boilers, or form separate communications, and the former is doubtless preferable, where the steam space is contracted.

The door for access, to inspect or repair any portion within the central smoke box, is on the uptake, when the latter is centrally situated, and on the flue alluded to when the same is introduced. When transverse space is available, the door is between the boilers.

The action of the flame within the boiler illustrated next demands notice. It may be

added, however, that although allusion will be made to the proceed from one grate, the same remarks apply to those portions on each side of the central smoke box.

The flame, on rising from the fuel on the grate, operates on the crown and sides of the fire box as previously described for other arrangements. On entering the combustion chamber a natural division ensues in the form of curves due to the position of the tubes. Now, the form assumed by the flame, and the action of the same on the back plate, depends entirely on the width of the chamber and amount of draught. With a maximum quantity of air, there is not the least doubt that the flame is accelerated in its progress, and will thus operate to a certain extent on the portions of the plate *opposite the bridge:* but the proceed from the grate, it must be remembered, enters the chamber in a bulk, its natural line of progression being to ascend: and the tubes are at right angles to the chamber, therefore the flame must recoil from the back plate, to a certain extent, before it can enter the tubes.

With an ordinary draught the velocity of the flame is greatly reduced; the effect, therefore, is, that the flame on entering the chamber in question simultaneously divides itself into two volumes, each proceeding in opposite directions. The portion of the back plate opposite the division, in this instance, loses much of the flame's action; but that opposite the tube plates is not so much affected.

The portions next operated on are the tube plates. The flame has now to alter its form into a series of projections from the bulk in the chamber:—further allusion is made to this when considering the effect.

The end smoke boxes are now supposed to be reached by the discharged flame from the tubes, and in these compartments an entirely reverse action ensues; because the flame, on leaving the tubes alluded to, expands into a *bulk,* and therefore an almost simultaneous rechange of form ensues to enter the, wing or, outer tubes. Another fact also is certain: the flame acts with greater force on the wing portion of the plate than that nearest the fire box. The line of progression is doubtless that of a curve, the form of which will be affected by the causes previously alluded to.

The final smoke box only remains to be explained; within this compartment four currents of gases are simultaneously received.

It is noticed that each of the wing tubes are opposite each other in duplicate arrangement. The uptake, it must be remembered, is at the end of the smoke box; the natural flow of the flame and smoke being towards that locality. Now, the proceed from the tubes nearest the final discharge opening, has less friction than from those at the hull side of the shell. This is obvious on consulting the plan, where the relative distances are seen. Apart from those facts, the forms assumed by the proceed from the two sources are worthy of comment. The flame emitted from the tubes at the hull side, proceeds horizontally and vertically, but that from the tubes at the uptake side conforms to an ascending motion only.

Thus far the action of the flame is expended, and the problem next requiring solution is the effect.

The action of the flame in the fire box being of a direct contact with the crown; and sliding on the sides, above the grate, the relative values

are obvious. But the effect in the combustion chamber forms a striking contrast with the previous conclusions alluded to. To return to the plan illustrated by Fig. 16—page 109—will greatly assist conception in the present instance. It is noticed that the bridge is opposite that portion of the plate virtually receiving the impact of the flame from the fire box. The natural effect of the bridge is, to cause the flame to undulate into the combustion chamber, the succeeding action being to fill that compartment to the extent permitted by the proportions of the supply and discharge.

The back portion receives the flame under the most disadvantageous mode of contact. This fact is not, however, wrong, but rather that it must be noticed when deducing the correct conclusions. The flame, when in the chamber in question, naturally inclines towards the alloted means of exit, *i. e.* the tubes. It is obvious that the line of progression is twin semi-circular, from the centre of the bridge to that of the disposition of the tubes. The flame being thus divided, that portion opposite the apex of the division is doubtless the least in contact, while the ends of the plate in question are likwise affected. Now if the acceleration of the flame is increased by an admission of air above or below the grate, the natural result is, that the area of the portion, not in actual contact, is decreased to that when the draught is lessened. The areas of the tubes also greatly determine the operation, or action, of the flame on the back plate. With a contracted discharge from the chamber in question the draught will be greatly retarded, and likewise combustion.

The cubic contents of the chamber is due to its width, height, and length, the two latter being determined from due causes, such as the number of the tubes, and height of the fire box. The space, imperative, between the tube and back plates is that required for repair, hence a fixed distance is the result in most examples. It is obvious that with a narrow chamber the flame acts on the back plate with a compelled contact, while the natural flow is to leave that portion. With the present as with the other examples, the crown is mostly operated on, and therefore the effect of evaporation on that part is the greatest. The value of the back plate, from those conclusions, is the same as for the last arrangement.

Next in value, of a higher ratio, is the tube plate: this is due to the tendency of the flame to act on the same, the line of progression being at right angles to the surface exposed, and thus causing a direct contact.

Now various opinions are entertained and put forth as to the relative value of the tube plate, therefore a little digression from the description will not be out of place. It is argued by some authorities that the perforations in the plate in question receive the direct flow of the flame: this is accounted for from the following causes.

The bulk of the volume conforms to the required form of tubular entry, before arriving at the plate. This is due to the fact that the spaces between the tubes actually *displace* the flame. Those spaces also are presumed to form the bases of cones, the apices of which are at the commencement of the division or alteration of the flame. The cones are of course not presumed to be circular at the base, but rather of an octagonal shape, thus conforming

to the relative position of the tubes. The "cones" are also imagined to be hollow at the base, therefore the actual contact of the flame with the plate, can be only at the connection of the hollows, also, it is certain that as this theory is correct, the plate in question is of inferior value as an evaporative agent.

Other authorities who do not accept the conical form as correct, prefer the assumption that the displacement produces hemispherical hollows; and this conclusion is undoubtedly correct in some cases, and equally conformable with natural laws.

The plate spaces between the tube orifices, it must be remembered, are bafflers in the accepted meaning of the term, because the flame does not voluntarily act on the tube plate, but is impelled towards the same.

Now, the natural flow of gases—as before stated—is curvous, and therefore the form of the flame between the tubes will be due to that law: therefore to cause the flame to assume an angle, a second surface is imperative to receive the first contact, the recoil producing the angular motion. The tube plate is a perforated surface, hence it is evident that with all forces of draught cones must be formed. And next it is requisite to consider what is contained in the cones. If it is gases, are they dormant, and how retained? If in a state of motion, whence the supply and discharge, and how the re-formation? It has been presumed that a vacuum is formed, but on reflection this is so contradictory in itself, that it is needless to explain away the fallacy.

The *actual* behaviour of the flame in front of, and on the surface in question, is due to many causes to which due attention will now be given. The flame first in contact with the interspersions disturbs the succeeding volume. A continual supply being maintained, a constant ebullition is the result. It is thus understood that a continuous displacement occurs before the bulk of the flame enters the tubes.

The draught impels the flame towards the nearest exit in all examples of arrangement, therefore, in the present, as in the other cases, the tube openings are the discharge passages.

Now, the flame—as before stated—must conform to the required shape before it can enter the tubes. Here a disturbance of the flame ensues in the form of continuous displacement against the plate, and the greater the velocity, the greater will be the *impact.* Hollow portions of flame are doubtless formed, but they are perishable, and constantly replaced by the succeeding current. The reason why these hollows are presumed to be curvous is simply that, the flame that acts against the plate *enters* the tubes—not rests on the interspersions—and therefore proceeds by forming curves at the edges of the tubes. The description here given readily pourtrays the fact, that natural laws regulate the motion of the flame from the combustion chamber to the tubes, as much as from the uptake to the chimney. It is obvious, also, that minute curved hollows are formed and destroyed in succession, with sufficient velocity to ensure the actual contact of the flame on the surface between the tubes.

The amount of draught and proportion of the tube orifices to the grate surface, will greatly affect the amount of contact. The compression of the hollows, or disturbance of the same, can only be effected by force of

draught and bulk of flame, which cause continuous displacement or ebullition.

The value of the tube plates of the combustion chamber now under notice, is as that previously alluded to, the position relative to the grate being the same in principle.

The ends of that chamber are both of equal value, being alike exposed to the flame's action. They are, however, of increased power to that of the last arrangement, being apart from the bridge. The values given relating to the tubes of prior arrangements, apply in the present instance.

The flame may now be said to be in the end smoke boxes. Here the effect of a return action ensues. The orifices receiving the flame, and the portion surrounding, are operated on precisely as those in the combustion chamber, but the value is duly lessened, due to the reduced temperature of the caloric.

The actual effect of the difference is very great, from natural causes. Water surrounding or supported on surfaces exposed to high temperatures, absorbs more of the heat in a given time than with lower temperatures. The penetration of heat increases in proportion to the rising of the temperature, therefore two laws are in operation to produce the required effect of evaporation.

The value of the portion of the tube plate receiving the flame is about one-third of that in the combustion chamber.

The tubes conveying the gases from the tube plate to the final smoke box, are operated on by the flame as those nearest the fire box, but the value of the former is reduced to about one-fourth.

The complete duty of the effect of the flame in the arrangement under notice is now concluded, and the next step is a summary of the results, arrived at, of the relative values of the various heating surfaces.

		Value.
INTERMEDIATE TUBES.		
Total surface	... ...	2·976
WING TUBES.		
Total surface	... ...	·744
FIRE BOX.		
Crown ...	Total surface ...	1·000
Sides ...	Above fire grate ...	·500
COMBUSTION CHAMBER.		
Crown ...	... ...	1·000
Tube plate...	Effective surface ...	·875
Back ...	Total surface ...	·700
Ends ...	Total surface ...	·300
END SMOKE BOX.		
Tube plate	{ Surface surrounding "wing tubes" } ...	·291

The principal advantages gained with this arrangement are, an equality of surface on each side of the main source, *i.e.*, the fire box, and the increased area of the tubular surface within a given space. Accessibility for inspection, repair, and renewal, are carefully considered and provided, so that practicability is combined with correct distribution of the details.

DIRECT RETURN TUBULAR ARRANGEMENT.

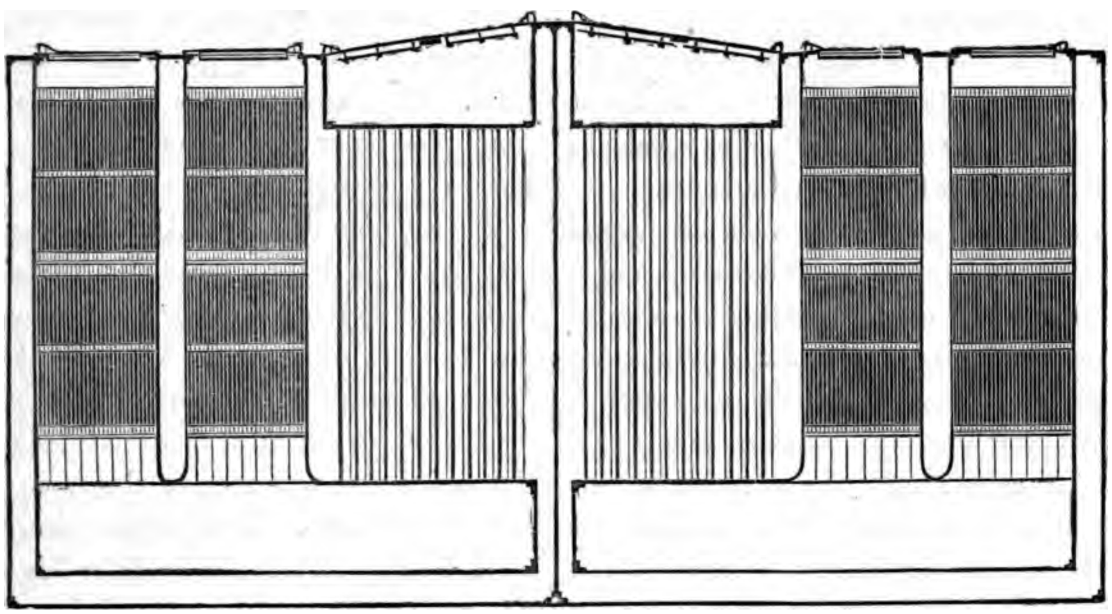
The illustrations, Figs. 15 and 16—pages 103 and 109—represent two arrangements especially designed for stoking fore and aft. Now, as before stated, one arrangement of the

low boiler *cannot* be universal, because the space allotted for a boiler not only determines the arrangement, but it also affects the proportions, which alteration, if possible, should be avoided; and the requisite horse power being given, all the heating surfaces should be of correct ratio to those of the grates; therefore the principal points to be noticed are: sufficient room to produce the caloric, and the most effective expenditure of the same.

Not only are the form and dimensions of the shell of the boiler deduced from the length and breadth of the space allotted, but room must also be preserved for stoking, cleansing, and renewing the tubes, therefore sufficient *length* in front of the respective localities is required. The positions of the trimming channels, to convey the coals from the back bunkers, also demand attention; and those considerations, together with those previously mentioned, constitute the main preliminary conclusions relating to the boiler room.

The following illustration—Fig. 17—represents a low boiler, designed for certain acquisitions. Transverse space in the hull is, in this instance, more available than longitudinally, being reverse to that with the previous arrangements. A plan only is given in the present case, the elevation being of such a simple character that it is obvious from what has been illustrated.

Fig. 17.



MESSRS. MAUDSLAY'S RETURN TUBULAR MARINE LOW BOILER.

The grates are two to each cluster of tubes, nearest the outer side of the shell. The combustion chamber is at right angles with the grate extending transversely of the shell, minus the water spaces. The tubes are arranged centrally between the fire boxes, at the sides of the same, thus producing a return action; the smoke boxes being at the boiler front, between the inner fire boxes.

Now, it is obvious that an increased length

of tubular surface can be obtained by securing the smoke boxes outside the front, rather than inside as shown:—sufficient space for cleansing and removal of the tubes would be available in that instance, due to the position of the fire boxes. The tubes, as shown, are—approximately—the same length as the fire bars; therefore what space is required in the one instance is convenient for the other.

The divisional or central plate separates the tubes, so that, in the event of mishap or destruction in either compartment, that adjoining is unaffected.

The inner sides of the smoke boxes join above the level of the upper row of the tubes, in the form of a semi-circle; the uptake is therefore common to both smoke boxes.

The action of the flame within the several compartments is the next consideration, and to make that matter clear, a comparative allusion is introduced:—the arrangement illustrated by Fig. 15—page 103—it will be remembered, has only one fire box, common to a single cluster of tubes; and the situation of the details at the forward end of the shell, is the same as the arrangement now under direct notice. But two distinct volumes emanate from separate sources, and proceed through a single discharge here; while with the former example the supply and the discharge are the same in number at one end.

Apart from this—for exemplification—it is stated in page 97, in allusion to the arrangement represented by Fig. 14—page 96—that "the line of actual contact is *not* entirely lost, due, of course, to the fact that the temperature of the flame emitted from the respective furnaces is unequal also in density and velocity." The same clause applies to the present arrangement, in as much that flames from separate sources *cannot cross* each other with certain arrangements.

The action of the flame in the respective fire boxes—Fig. 17—are about equal, the crowns and the sides above the grates are duly operated on, as in the previous examples.

The volume first to enter the combustion chamber, with the greatest velocity, is that emanating from the inside fire box, the proceed from the next grate being in due ratio. This is obvious, on considering the distance of the outer grate from the tube plate. Also, another cause is that the flame from the inner grate has less friction or surface in contact than the sister volume.

The situation of the uptake being central, it may be argued that the tubes nearest the division plate receive the greatest amount of the flame through them. The natural law—"that all gases proceed in a direct line,"—tends to impart the idea that the tubes nearest the inside fire will receive the flame direct from that source. Another fact also, before alluded to, again presents itself for due notice, "flames or gases with a natural flow do not *cross* each other." It is therefore obvious that the volume from the outer fire grate acts—passing beyond that from the inner—against the back of the combustion chamber, and virtually enters the tubes nearest the end of that compartment. It is not presumed that a separate flow in front of the tube plate is maintained, because, on the contrary, it is certain that time is permitted for amalgamation at that locality.

The action of the flame on the back of the

chamber is of a sliding nature, and the curves formed by the flame—on entering and discharging—reduce the area in actual contact. The ends are also duly affected. From those conclusions the values of the crown, back, ends and tube plate are as the example previously alluded to.

The action of the flame within the tubes is as that with other arrangements, and the surface will therefore be of equivalent value.

The surfaces of the smoke box cannot be considered worthy of notice for evaporation, although it is probable that the temperature of the flame emitted from the tubes is not sufficiently absorbed during its traverse through the same to warrant that entirely.

From this cause the remarks made that "the tubes could be practicably longer in the present arrangement," applies. The objections, to the projected smoke box, are of course the requisition of right and left hand stoking, and loss of bottom water spaces. These, however, are matters of minor consideration, in proportion to the gain of evaporative surface.

The following table of the relative values of the surfaces now described will be found of utility when comparing the proportions of the various arrangements :—

		Value.
TUBES.		
Total surface...	... ...	2·976
FIRE BOX.		
Crown	... Total surface ...	1·000
Sides	... Above grate ...	·500
COMBUSTION CHAMBER.		
Crown	... Total surface ...	1·000
Tube plate	... Effective surface ...	·875
Back plate	... Total surface ...	·700
Ends	... „ „ ...	·250

These numeral conclusions of course refer proportionately to their special whole numbers, the application of which will be obvious, when calculating the relative amount of evaporation.

ANGULAR GRATE ARRANGEMENT.

The title of the boiler now under consecutive description, is somewhat singularly adverse to that relating to Fig. 14, page 96. There the tubes can be angularly situated while, in the present case, the grates assume that position, in relation to the remaining details.

Enough has been said to conclude that transverse space in the hull is one of the prior considerations when arranging the low boiler. And with the present example illustrated, in plan only, by Fig. 18—page 118—the fire grates are at given *angles* to the longitudinal line of the hull for that purpose; and is especially designed for available space for stoking between the two shells also.

Before proceeding further with the description, the *space* required for stoking, is, doubtless, worthy of comment.

The distance allotted, generally, between the high boilers, particularly for the government practice, is about ten feet. With this distance, ample space is available, when stoking, to pass in front of the opposite boilers without fear of collision : and independently of the dimensions alluded to, consideration must be given to the frontage requisite, proportionately to the length of the grate surface.

The position of the stoker when agitating the fuel at the back, or bridge end of the grate, is within two feet of the boiler front. Now, the actual space occupied in front of the fire door when operating at or near the dead plate, is

about one foot to eighteen inches longer than the grate surface. From this fact it is obvious that the length of the space, from the front of the boiler to the bridge of the fire box, determines the frontage requisite.

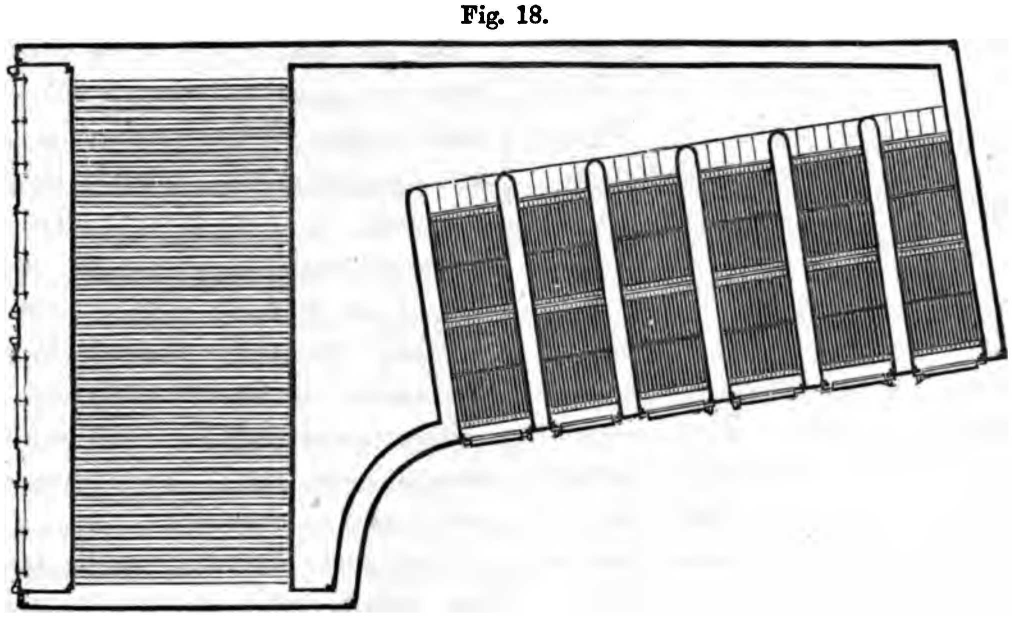
The number of the furnaces in the present arrangement exceeds that of any yet alluded to, the width of the tube arrangement being of similar ratio. It is apparent, also, that the width of the furnaces, in proportion to their length, in the present instance, is much less than with the prior examples.

Now, as to the better proportion. Many authorities introduce in their practice a length of six feet six inches for the grate, and a width of two feet eight inches to three feet. Other experienced members of the profession prefer a width of two feet six inches as a maximum, with the omission of the fraction to produce the minimum.

To decide hastily in any instance of comparison, often refutes actual results; therefore, with the present matter at issue, a brief consideration of the requisition is given.

The principal aim of the designers of marine boilers, should be to produce the greatest amount of evaporation, with the least amount of space, material, and consumption of fuel. The area of the grate surface is, of course, the first consideration before deducing the remaining proportions. Now, the area, in the strict meaning of the term, is simply conventional. It must be remembered, that the width has much to do with the combustion effected. The height, also, greatly regulates the temperature attainable. Some authorities attach a great deal of importance to the contact of the red hot fuel with the sides of the fire box, thus advocating the narrower grates; and it has been known in some cases, that one-third of the evaporation actually proceeds from the surfaces in direct contact with the solid fuel; therefore the tem-

Fig. 18.



MESSRS. MAUDSLAY'S ANGULAR GRATE MARINE LOW BOILER.

perature of the fire is the next consideration before concluding as to the correct proportions of the fire box.

According to Professor Rankine, the temperature of the products of combustion is known from the formula: "Total heat of combustion of one pound of fuel, divided by the weight of the air and fuel, and the specific heat of the whole products of combustion."

Now as this formula may seem perplexing to some, the following description is given. The products of combustion of coal are, carbonic acid gas, nitrogen, air, ashes, and steam; the specific heat under constant pressure of each being—

	Value.
Carbonic acid gas	·217
Nitrogen	·245
Air	·238
Ashes	·200
Steam	·475

The actual mean is ·275, but to ensure correctness or allowance for any portion that may not be as its equivalent numeral, the "recognised" mean "specific heat" is assumed as ·237, being ·037 less than the "actual," and ·001 less than the *air*.

According to the authorities on this subject, the total heat in one pound of carbon is represented by 14,500 units, if completely burnt so as to make "carbonic acid." The weight of air, as before stated, required per lb. of fuel is 12lbs. From these numbers and the cause of their presence, the matter resolves itself into simple calculation.

Now, presume it is required to know the highest temperature a furnace can give out,

$$\frac{14500}{12+1 \times \cdot 237} = 4706^\circ \text{ Fahr.}$$

If air is introduced to a greater extent than what is specified, the temperature will be lowered in due proportion. For example, presume the amount of air per lb. $= 19 + 1 = 20$, thus, $\frac{14500}{20 \times \cdot 237} = 3059^\circ$ Fahr., which is doubtless the maximum in practice. The consideration, of the amount of air required to consume one lb. of fuel, is of the highest importance. The velocity of the air is also worthy of notice, being in due ratio to the rate of the consumption of the fuel and cubic contents of the furnace. Professor Rankine has calculated the following table, which imparts much information within a small compass:—

Temperature of Furnace. FAHR.	Supply of air in lbs. per lb. of fuel. 12	18	24
	Cubic capacity in feet of gases or air per lb. of fuel.		
4640°	1551		
3275°	1136	1704	
2500°	906	1359	1812
1832°	697	1046	1395
1472°	588	882	1176
1112°	479	718	957
752°	369	553	738
572°	314	471	628
392°	259	389	519
212°	205	307	409
104°	172	258	344
68°	161	241	322
32°	150	225	300

Here a special fact is arrived at, viz., the more air admitted above 12lbs., the lower the temperature, the velocity of the gases being duly noticed. It also seems certain that the admission of air to the grate to a greater extent than the 12lbs. or 906 cubic feet per lb. of fuel consumed with a temperature of 2500°, is injudicious to effect economy.

Now the temperature of cast iron fully melted is 2754°, 2786°, and 3479°, all per Fahr., by three authors on that subject; the mean of these results is 3006°, which may be taken for the present purpose as correct. It is therefore apparent that it is possible to cause a furnace to impart heat of a higher temperature than melted iron. How far this attainment is practicable is only due to the conducting powers of the material between the flame and the fluid. Hence the result that the plates forming furnaces often *carbonize* to the requisite *thinness* to maintain the requisite *conduction*.

The presumed gain with narrow furnaces, in relation to the admission of surplus air, is that a given volume can only enter into a proportionate space. For example: a given area of grate surface divided by the length = the total width, which divided by the number of furnaces = the width of each furnace. The amount of air admitted during the operation of stoking is in due ratio to the area above the grate, hence with narrow furnaces the admission is the least. Again, for example, six furnaces, each 2 feet wide = 12 feet, and $\frac{12}{4} = 3$ feet, which will be the width of each for four furnaces. It is obvious, therefore, that with the latter width, $\frac{1}{12}$ amount of air will be admitted during stoking, more than with the former, always remembering that the doors are opened and closed consecutively.

Now, in allusion to the idea entertained by some, that one-third of the evaporation actually proceeds from "the surfaces in direct contact with the red hot solid fuel."

It has been proved in this work that the value of the side surface is inferior proportionately to the crown. It is certain, also, that rapid combustion produces heat, and that the rate of combustion is in proportion to the correct distribution of the air. From these facts it is obvious that the sides of a narrow furnace can be of no more value than a wide one, and, independently of the number, there does not seem to be much gain in their introduction—admission of air excepted. Other objections consist of contracted space for stoking, and the small amount of fuel congregated.

To maintain a given amount of evaporation continuously, a proportionate temperature is requisite, and, therefore, the quantity of fuel imperative. It is obvious, also, that the ratio of combustion exceeds that of the relative quantities of the fuel, *i.e.*, a grate of small but correct dimensions will impart more heat in proportion to the fuel consumed, than one incorrect. This is more apparent on remembering that combustion is the amalgamation of gases, and the more perfect the mixture, the higher the temperature as the result. From these conclusions it is certain that two feet six inches to three feet—being the general practice—is the correct width for a marine fire grate, the length being subservient to the required area.

The description of the arrangement of the boiler, now under direct notice, Fig. 18—page 118—must next engage attention. The fire grates are angularly situated in the shell—at right angles—with the front. Now the cause for this angle of position is to admit the requisite area in the combustion chamber, for the exit of the flame after passing the bridge.

The combustion chamber is peculiarly formed, not unlike that for the "right angle tube" arrangement.

An exception to the similarity, however, presents itself for comment, in relation to the details. With the right angle tube arrangement the fire grates are in a line with the portion of the combustion chamber in front of the tubes; but with the arrangement under notice, the grates are nearly at right angles with the tube plate, hence the comparison is an extreme contrast.

The combustion chamber also, it is noticed, is apparently formed to be common to two requirements, *i.e.,* combustion and repair. In the first case, space for the amalgamation and discharge of the gases is considered; and in the second, room for the necessary manipulation is duly provided.

The arrangement of the tubes is a single cluster, without intervention. The smoke box and uptake are situated at the extremity of the shell of the boiler, ample space being allotted for cleansing and renewing the tubes.

As only the sectional plan is illustrated, a description of the elevations will not be out of place. Assume a sectional elevation, longitudinally of the shell: the fire boxes are curved at the crown and base, the sides being parallel; while, in some examples, the fire boxes are cylindrical, to ensure greater strength. The sectional form of the combustion chamber is a parallelogram, with rounded corners to form the connection of the sides, crown, and bottom. The tubes, of course, present the ordinary form of connection between the plates. The smoke box and uptake are the usual kind, with doors and dampers for the respective purposes.

The longitudinal section through the fire box —or transversely of the shell—presents the ordinary arrangement of fire door, grate, and bridge, the inclination of the grate being the usual ratio to the horizontal line. The transverse section of the combustion chamber assumes a parallelogram, with the general mode or form of connection and jointing.

The next portion of the subject under notice is the action of the flame on and through the surfaces and spaces allotted. It is noticed that six fire grates are in operation at the same time, and the proceed of each passes through one final passage of exit.

It is next necessary to consider the action of each separate bulk of flame from the respective sources, but first must be noticed the situation of the uptake, before explaining the line of progression.

It has before been stated that all gases proceed in a direct line, also that flames or gases, with a natural flow, do not cross each other; and, in order to render these remarks obviously applicable in the present case, a brief definition is made.

The position of the uptake in the present example is at the centre of the longitudinal line of the hull, or between the shells of the boilers; and this position for the discharge, relative to the supply, governs the line of the flame's progression throughout its traverse.

Now the direct line from the extreme fire grate to the uptake crosses the remaining grates. Here the observation of the natural law—"flames do not cross each other"—becomes appropriate; also the distance from the uptake demands attention.

It is evident also that gases of the same density will amalgamate, and not displace each other under natural progression. It is well known too that two gases of separate densities,

the one less rarefied than the other, do displace each other, and the lighter does not penetrate through the heavier.

These facts having been duly noticed and commented on, to prove the actual action of the flame in the present arrangement, the following practical description is given.

Assume that the six grates are alike supplied with fuel, and that the temperature is raised sufficiently to produce rapid combustion,—it being remembered that any notice of the action of the flame before a given temperature is attained, or the boiler producing steam, will not be applicable for the present purpose.

The proceed from the fire grate nearest the uptake is the first to enter the tubes similarly situated; the distance producing this cause as much as the locality. The action of the flame on passing the bridge is curvous, and theoretically, the line of progression from the extremity of the fire box to the tube plate is a perfect angle. But the deviation from the angular line is due to the succeeding volumes from the respective sources; also, as before stated, the denser gases must succumb—from the direct line of progression—to those of greater rarefaction. Were this not the result, the portion of the combustion chamber opposite the tube plate would be valueless as an evaporative agent; whereas, in practice, the contact of the flame is certain. There the action of the flame on entering the chamber is doubtless a curve, on the outside of the volume, the inside portion conforming to the duplicate sides of the fire grates and the curved and end portions connecting the tube plate.

The volume from the next grate, after leaving the bridge, passes in a direct line, theoretically, to the tubes, beyond that preceding it, and the same action occurs from each consecutive source, until reaching the fifth grate. Here, doubtless, a certain amount of amalgamation with the proceed from the sixth or last fire box ensues.

By this it will be understood that the amalgamation is due to the temperature of the combustion chamber, because the proceeds from the two extreme grates have to pass through the greatest distance from the uptake, hence time is permitted for rarefaction. Another cause: the proceeds from the four preceding grates are of a denser volume, each proportionately to the distance from the final discharge.

Now it may be argued that the rarefied volumes—due to the natural or direct flow—will pass through the most dense volume, and reach that portion of the tube plate most common to the uptake. Such, however, cannot be; and this is rendered fully apparent on considering again that "gases of separate densities do *not* cross each other;" and that the *distance* between the supply and the discharge determines the "line of progression."

The action of the flame on the plate in front of the bridges or fire boxes is of a sliding nature, combined with impact. The latter operation extending only for about one half of the plate's length—commencing at the fire grate end. The flame's action on the extremity of the chamber is sliding, the contact extending only above the bridge.

The tube plate receives the action of the flame angularly, rather than direct. This is, of course, due to the position of the grates, in relation to the plate alluded to. The flame,

after acting on the portion of the chamber in front of the grates, inclines towards the tube plate, in a line with the uptake; hence the angular contact with the surface in question is obvious.

The tubes retain the flame within them proportionately to their length and velocity of the draught, being similar in fact to other arrangements.

The flame has now been practically followed to the extent of its duty: and the problem to be solved next is its effect. Before, however, entering into that question, allusion must be made to the relation of the arrangement to the effect attainable. It is seen that the proceeds from the fire grates are common to one discharge before they enter the tubes, *i.e.*, the space between the back plate and the curved portion, at the bridge end of the grate nearest the tubes, is the area through which the entire volume passes. Now practical demonstration has long ago proved that the admission of air—above the grate—more than a certain quantity, lowers the temperature of the flame in due proportion. It is obvious, also, that the fire doors must occasionally be opened during the operation of stoking, and it is due to this latter requisition that the present arrangement is affected.

The arrangement, Fig. 14—page 96—has already been adverted to in allusion to the combustion chamber. The outside grate, in that case, is an independent source; but, in the present case, the entire proceed is common to one exit before entering the tubes.

Now, it may be argued that the various volumes on entering the combustion chamber amalgamate, and the presumed defect is a universal error. This cannot be answered better than by alluding to Figs. 12, 13, 15, and 16, on pages 83, 93, 103, and 109, where it is seen that an independent exit from the fire box is generally preserved, and the opening of either fire door only affects the grate directly connected. Apart from this, it must be remembered that air admitted into a single source—having an independent traverse—does not materially affect the remainder of the currents or volumes, due, of course, to the density of the gases. It has been stated before, also, that the amalgamation of gases is only certain when the same are common with each other, in weight, bulk, and property.

Now, to further digress from the descriptive portion of the effect of the flame in the arrangement now under direct notice—Fig. 18, page 118. Presume the door of the furnace, at the extremity of the shell, is opened for the purpose of stoking or otherwise: the extra air admitted for the given time will not only affect the volume in that fire box, but will follow the preceding currents from the different sources.

On closing the door alluded to, and opening that nearest the tubes, the action is reversed, the volumes from the succeeding grates meet the cooler gases; and if an intermediate door only is opened, the cooler gases are confined between two volumes of simultaneous progress.

Now as to the general effect of the flame. The crowns and sides of each furnace are doubtless proportionately alike operated on by the flame. If any variation, the fire box at the tube plate end of the chamber is perhaps the least effective, due to time and distance. The end of the combustion chamber

is affected by the flame in an inverse ratio, *i.e.*, the greater the draught, the more the tendency of the flame to leave the surface in question. The portion of the surface in question below the bridge is not worthy of much notice as an evaporative agent; but conclusions on this point have already been explained, and therefore need no further comment now.

The back of the chamber is the next surface to be noticed. The flame affects this portion more or less entirely, the least portions operated on being at or near the ends, due of course to the fact that the line of the flame's progression is curvous at the extremities.

The portion of the chamber in front of the tube plate, and the end connecting the same, is exposed to the same effect as the back plate of ordinary combustion chambers. The flame is impelled from this surface rather than towards the same. In the present instance, the curved portion doubtless is mostly operated on, due to the contracted space at the connection with tube plate. The end is doubtless of the least in value, there being no cause for the flame to act on that part with any effect conducive to evaporation. The crown surface of the combustion chamber possibly receives the greatest effect of the flame, there being two operations in action at the same time, viz., a sliding and almost direct contact.

Next, as to the effect of the flame on the tube plate. Before, however, deducing the same, allusion will be made to the natural line of impact.

In page 113, a description is given of the actual "action of the flame on the tube plate." In allusion to the "hollows" formed by the interspersions, it is stated that, "The compression of the hollows, or disturbance of the same, can only be effected by force of draught and bulk of flame, which cause continuous *displacement* or *ebullition*." From that description it is certain that impact is the acquisition to render the tube plate of raised value as an evaporative agent. The line of the flame's progression has met with due notice. Attention must now be given to the line of impact.

With the high boiler and the return tubular low boiler, the lines are alike in form, the arrangement of the details being the same in principle. This is further rendered obvious on assuming the side of the shell of the low boiler to be the base line of the high. The line of impact, in these instances, forms two right angles with return extremities, the one from the grate and the other from the combustion chamber.

The form assumed with the present arrangement now under notice, is composed of two extremities at right angles, connected by a third line forming an obtuse angle, or acute, as the draught may determine.

Now it may be argued that, with the return tubular arrangement, the flame has to follow a reverse course; hence the value of the tube plate, in that instance, must be duly lowered in proportion to the present disposition. On due examination and recognition of natural laws, it will be understood that the distance of the tube plate from the bridge, and the resistance to which the volume is exposed, has much to do with the impact of the flame on the tube plate.

It is a known law, in mechanics, that "all points of contact are mostly affected when the line of impact is at right angles with the sur-

face operated on." Now, happily, nature does not give out capricious ideas in any instance. Wonderfully true is the fact that she has but one law for all things, whether pertinent to fluids or solids. To understand this is the aim of all scientific aspirants; and certain it is that, the deeper the research, the less the self opinion of prior knowledge. Designs and schemes that were once deemed perfect, are thus often gladly thrust aside to make room for those more akin to the simplicity and truth of natural laws.

The vital attainment with the marine boiler is the contact of the flame with the surface exposed. This is certain with the crowns of the furnaces and combustion chambers, natural laws being under no restraint.

In the case of the tube plate, the flame has a tendency to escape from the same; and were it not for the bulk of the volume, the surface in question would be of little or no value for evaporation. The line of the flame's progression, in front of the tube plate, is generally either angular or direct—rarely of a sliding contact.

Could the latter be attained, it might be argued that it would doubtless produce the greatest effect, time being permitted for absorption of the caloric, it being certain that the flame must escape at right angles. This, however, being contrary to the fact that impact produces contact, the effect is the least, as the flame when sliding actually rises from the surface before entering the orifices.

Now, when the flame progresses towards the surface in question, angularly, the contact is increased to that when sliding; and the more obtuse the angle, the greater the effect. Therefore when the volume from the combustion chamber proceeds direct against the tube plate, the contact is the greatest; due, of course, to the impact being more effective.

In order to render these facts more obvious, the following descriptive illustration is given, remembering that nature has but one law.

Presume a surface of an even plane is exposed to a blow, or series of blows, from a body, separate from the cause of impulsion. It is requisite to produce the greatest effect with the least cause. The power exerted is to be expended at the point of contact, or at the termination of the blow. It is almost needless to state that the line of progression or propulsion must be at right angles with the surface or plane line, to attain the acquisition.

If the seat of power and line of progression is at an angle with the surface operated on, the impelled body will strike the same angularly with less force, although the exertion may be the same as before.

It may now be wondered what has become of the power; it is stated to be the same as in the first experiment, but the impact the least in this case. Here the truth of natural laws becomes wonderfully obvious. A certain force has been exerted, and it cannot be lost; there must be an expenditure equivalent to the supply; or to be concise, nothing is lost, but rather expended.

The body, after leaving the surface it struck obliquely, retained a given portion of the power, and the impact was therefore duly lessened. Were this not the case, the concussion would have been equal to that produced at right angles.

Again, for exemplification, presume the sur-

face alluded to is perforated—the orifices being sufficiently large to receive the separate body in question without friction or adhesion. Now it is presumed that it is requisite to impel the body, at a given distance, to enter either of the perforations. The same rule again applies as before: at right angles must be the line of progression. This is more imperative if the perforations are prolongations or similar to the boiler tubes. To add more remarks on these points is superfluous, and the cause for those introduced is simply to prove that nature should be consulted in every branch of science.

The effect of the flame on the tube plate in the arrangement now under direct notice is less than in the preceding examples. This is due to the position of the grates and uptake causing the line of impact to be angular.

The tubes are not of lower evaporative power in this example than in others alluded to, owing to the traverse of the flame being the same in each instance.

Now, as to the value of the heating surfaces alluded to. The crowns and sides of the fire boxes are the same as before determined. Of the same relative value, also, is the crown of the combustion chamber, although the surface is considered to be unequally operated on by the flame; and considering the different temperatures, the conclusion given is doubtless correct.

The back of the combustion chamber is considered to be of a higher ratio than any preceding it, and the reason for this conclusion being considered a reality, is simply due to the arrangement of that chamber. The proceed from each furnace tends to impel or displace that from the preceding source, and the result is that, from the natural resistance, contact of flame is produced. The portions below the bridges, and at the ends, are of course relatively operated on; but, considering the entire surface in proportion to others alluded to, there is a gain in the evaporative value.

The value of the angular and curved portion in front of the tubes is not, of course, equal to the back—due to the relative situation. The flame—as before stated—always tends to recede from rather than act on the surface in question; its value therefore is proportionately lessened. Enough has been said in relation to the tube plate to conclude its value, which is less than the ordinary ratio previously given.

The values of the ends of the chamber are considered equal to that of the last arrangement. The tubes also are alike deemed to give out the same evaporative result.

The following table renders the conclusions arrived at of concise recognition:—

TUBES.			
Total surface		... ...	2·976
FIRE BOX.			
Crown ...	Total surface	...	1·000
Sides ...	Above grate	...	·500
COMBUSTION CHAMBER.			
Crown ...	Total surface	...	1·000
Tube plate ...	Effective surface	...	·800
Back plate in front of furnaces	Total surface	...	·750
Side plate in front of tubes	Total surface	...	·600
Ends ...	Total surface	...	·250

It is almost needless to add that these decimal numerals are in direct ratio to their respective whole numbers.

DIRECT TUBULAR ARRANGEMENT.

The design of a marine low boiler, as before stated, is always subservient to its destination, but apart from this repair must be remembered.

An arrangement set out on paper often presents the most inviting aspect for construction. The attainment of heating surface in the smallest amount of space has been, perhaps, duly attended to. The height of the shell is reduced to the minimum above the crown of the fire box. The tubes are arranged in the most fanciful disposition, and altogether the entire arrangement is deemed a success as far as the contraction of cubical contents is requisite.

Apart from these attainments, imagine that space for repair has not been attended to. The result is, that when inspection and renewal become imperative, what was once deemed a gain will be condemned as a loss.

Now to still further illustrate the correctness of the conclusion alluded to. Let it be presumed that a low boiler is designed for an allotted locality of the most stringent requisition. The amount of evaporation to be a fixed result.

The plans and elevations of the shell are of course the primary illustrations formed by the designer. The fire boxes, combustion chamber, tubes, smoke box, and uptake, follow in due order. Assume also, that the most intricate arrangement is decided on to attain the requisite amount of heating surface.

The boiler is constructed, and the imaginations become realities. Also when under steam, the result is equal to the expectations, and the designer congratulates himself on the success of his power of thought.

The boiler for a given time possesses ample means of producing the requirements, but after a while cleansing and repair become imperative. Then only does the *practicability* of the arrangement become tested. Then, perhaps, it is seen that the space allotted for the passage of the flame is insufficient for the workmen to perform their duty. Or worse still, one portion of the internal part must be removed to repair that requiring the same. Further than this, perhaps the stays are so numerous above the fire boxes and tubes, that practical inspection requires their prior removal.

All these preliminary operations, to attain access for repair, lower the value of the design in question. The truth then presents itself, and the practical lesson is acquired, that access for repair demands as much consideration as compactness of arrangement.

The arrangement of low boiler, illustrated in sectional plan by Fig. 19, on page 128, is especially adapted for longitudinal space in the hull. The fire boxes are separate compartments, common to one combustion chamber, of the simplest form. The tube plate is at right angles with the fire boxes, and the tubes continued in a line with the latter. The smoke box is at the extremity of the shell, parallel with the combustion chamber.

From this description the longitudinal view is understood to be simply a direct position for the details, *viz.*, fire box, combustion chamber, tubes, and smoke box. A transverse section through the fire boxes will present the tubes appearing over the bridges in a line with the grates.

It will, doubtless, be obvious that the arrangement now under notice is common to simplicity of design and access for repair. The attainment of these two relative functions is especially aimed at in this instance, as much as the hull space for the reception of theshell.

Before proceeding to treat of the action of the flame, it will not be out of place to allude to the strength of the arrangement under direct notice.

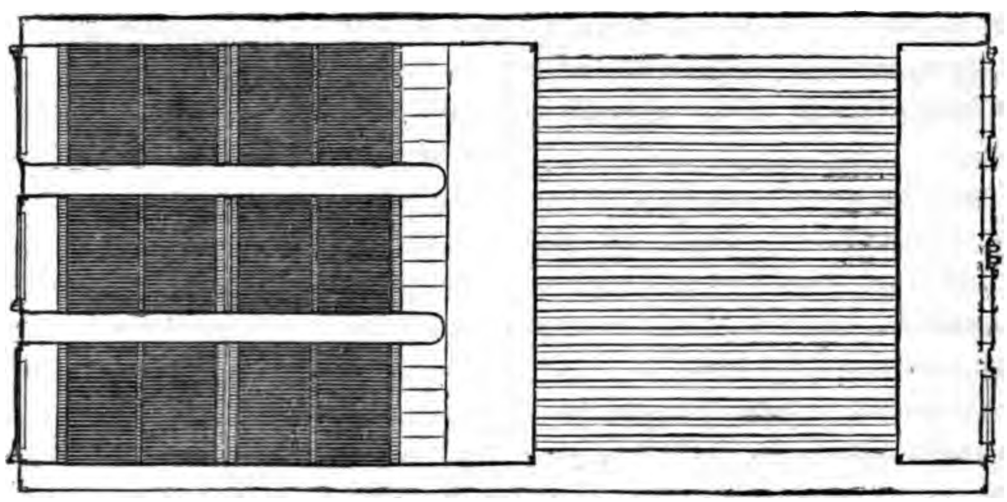
Now a tube-like form offers the greatest resistance of all. Nature herself teaches this in all her productions—for example, the most fragile tubular members of the vegetable kingdom are the strongest, proportionately to the sectional area of the solid kind. The elder, the reed, the wheat-stalk, and divers other tubular-formed plants, all bear evidence of the supreme wisdom of the great Creator.

Now it may be argued that, if the cylindrical boiler possess these merits, which nature herself bestows, what is the cause of its not being more generally adopted for marine purposes? Loss of space seems to be the greatest barrier. This is obvious on remembering that when a series of cylindrical shells are laid side by side, a useless vacancy is formed above and below the centre line; and it is evident that this loss of space in the hull causes the introduction of the flat or square-shaped boilers. The latter can be packed in a given space without waste of area or cubic contents.

It will perhaps be as well to allude to the strength of cylindrical boilers, from actual experiment and practice, before further alluding to the description.

W. Fairbairn, Esq., C.E., F.R.S., &c., treating on the "strength of boilers," observes:—

Fig. 19.



MESSRS. MAUDSLAY'S CYLINDRICAL MARINE LOW BOILER.

"To make a sound and perfect boiler—or as near perfection as possible—we have to consider the nature and working pressure of the steam; the best form of boiler necessary to resist that pressure; the maximum of safety; the properties of the material, the union of its parts, and the condition under which it should be placed to produce satisfactory results; and, boiler explosions.

"I have already shown that the cylindrical boiler is the only form calculated to resist the

elastic force of steam, and that the greatest care is necessary to be observed, not only as regards the strength of the plates, which should be of the best quality, equal to a tensile strain of twenty-one tons per square inch, but they should be double riveted, if we are to have a perfectly strong and well constructed boiler. But before we enter upon the art of construction, we must devote attention to some of the properties of steam as regards its temperature, pressure, volume, and density, and we shall then have a more clear conception of the forces with which we have to deal, and how to regulate these forces, and construct vessels to retain them without risk to property or any of those casualties which endanger life.

"TABLE I.

"Table of Temperatures, Volumes, Pressure, &c.

Pressure per Square Inch in lbs.		Corresponding Temperature in Fahrenheit.	Relative Volume of Steam compared to Volume of Water that produced it.
Below the Atmosphere	1	102·9	20954
	5	161·4	4624
	10	192·4	2427
	15	213·0	1669
Above the Atmosphere	1	216·4	1572
	5	228·3	1280
	10	240·7	1042
	15	251·2	882
	20	260·3	765
	22	263·7	727
	30	275·7	608
	50	299·1	434
	70	317·8	340
	90	332·2	281
	105	343·3	249
	135	360·8	203
	165	375·6	173
	180	382·3	161
	210	394·6	141
	225	400·2	133

"Now, as these forces have to be retained within comparatively small limits, we must endeavour to ascertain the force which tends to rupture a cylindrical boiler in the direction of its axis, or to separate the ends from the sides. To accomplish this we have only to multiply the area of the ends in inches by the number of units of force applied to each superficial inch, and the result is the total divellent force in that direction. To resist this, we have the area or number of square inches of the plates in the circumference, as a counteracting force, which acting by tension, will retain the ends in their places so long as the strength of the iron or the riveted joints exceeds that of the internal force, or until the moment of rupture, when they become equal. Let us, for example, suppose a boiler of six feet diameter and thirty feet long, to be composed of three-eighth inch plates, whose ultimate strength is twenty-one tons per square inch, and we have with steam of sixty pounds pressure, a force against each end of the boiler of 244,290lbs. = 109 tons. To this force we have a resistance equivalent to the area of the plates $84{\cdot}75 \times 21 = 1779{\cdot}75$ tons, which gives a large margin of strength, being in the ratio of 1779 : 109, or nearly as 16 : 1.

"This excess of strength is evidently great, but I have already shown by direct experiment that we must not calculate upon such a powerful resistance as twenty-one tons per square inch, but must reduce it to the following standard, viz.:—

"If we take the ratio standard of the plate at 100, we must reduce it for double riveting to 70, and for single riveting to 56, so that we have the resistance in the ratio of the numbers, 100, 70, and 56. Now, as very few boilers

are double riveted—unless it be locomotives—we come to the standard of 56 instead of 100, and in the place of the boiler being equal in its powers of resistance to 1779·75 tons, as given above, it would burst with 996·6 tons, being in the ratio of 996·6 : 109, or in other words, it is nine times stronger than the assumed pressure at which it is worked. This is not, however, the case as regards the curved sides, which have a tendency to rupture along the whole length of the cylinder upon each lineal unit of its diameter. With the forces in the direction calculated to divide the cylinder in halves, the resistance would be represented by multiplying the diameter by the force exerted on each unit of surface, and the product by the length of the cylinder, which gives the divellent force in that direction.

"Taking the boiler which we have selected, 30 feet long and 6 feet diameter, and plates $\frac{3}{8}$ thick, and we again have

$$\frac{72 \times 60 \times 360}{2240} = 694 \text{ tons}$$

as the pressure acting upon both sides of the circumference throughout its whole length.

"Now, assuming that the plates with single riveted joints are equal in their powers of resistance to 34,000 lbs., or about 15 tons per square inch, we then have according to the above rule $\cdot375 \times 360 \times 2 \times 15 = 4050$ tons as the force that would burst the boiler. It has, however, been shown that the collective force upon the longitudinal seams is only 694 tons, consequently we have an excess of strength in the ratio of 4050 : 694 or as 6 : 1 nearly. Now, this is not too large a margin of security, but it is sufficient, provided the plates and workmanship are of the best quality, otherwise it would be desirable to have thicker plates. To this, however, I decidedly object, as there is no economy in the use of an inferior material: on the contrary, it is highly injurious as regards the transmission of heat, and not to be depended upon when composed of an inferior quality of iron. In every case of boiler construction it is essential that we should avoid the introduction of inferior plates, which in general partake more of the crystalline than the ductile character, and are therefore highly objectionable in the construction of boilers which have to resist so powerful an agent of destruction as the elastic force of steam.

"On this part of the subject I may advert to facts, that on referring to the comparative merits of the plates composing cylindrical vessels subjected to internal pressure, they will be found in this anomalous condition, that their strength in their longitudinal direction is twice that of the curvilinear direction. This appears by a comparison of the two forces, wherein we have shown that the ends of the three feet boiler, at 40 lbs. internal pressure, sustain 360 lbs. of longitudinal strain upon each inch of a plate a quarter of an inch thick; whereas plates of the same thickness have to bear in the curvilinear direction a strain of 720 lbs. This difference of strain is a difficulty not easily overcome, and all that we can accomplish in this case will be to exercise a sound judgment in crossing the joints, the quality of the workmanship, and the distribution of the material. For the attainment of these objects, the following table, which exhibits the proportionate strength

of cylindrical boilers, from 3 to 8 feet in diameter, may be useful.

"TABLE II.

"Table of Equal Strengths in the External Shell of Cylindrical Boilers from 3 to 8 feet diameter, showing the thickness of metal in each respectively, for a Bursting Pressure of 450 lbs. to the square inch.

Diameter of Boilers. Ft.	In.	Bursting Pressure equivalent to the Ultimate Strength of the Riveted Joint, as deduced from Experiment, 34,000 lbs. to the Square Inch.	Thickness of the Plates in Decimal Parts of an Inch.
3	0	450 lbs.	·250
3	6		·291
4	0		·333
4	6		·376
5	0		·416
5	6		·458
6	0		·500
6	6		·541
7	0		·583
7	6		·625
8	0		·666

"There is another question relating to the strength of boilers which requires careful attention, viz., the internal flues and their resistance to external uniform pressure. In calculating the strength of boilers the internal flues, until of late years, were never taken into account. They were always considered much stronger than the exterior shell, and that there was no danger from collapse. Yet in the very face of these conclusions numerous instances of fatal explosions occurred, not from the weakness of the boiler itself, but from collapse of the flues which, at a subsequent period, were found from actual experiment to be the weakest part of the construction.

"From the first commencement of boiler construction to a very recent date, we all of us acted under the impression that the flues were the strongest part of the boiler, and that a perfectly cylindrical tube, when subjected to a uniform pressure, converging upon its axis, was equal in its power of resistance, irrespective of its length. This was, however, an erroneous opinion, as I found, on submitting a series of cylindrical and elliptical tubes to external pressure, that they were weak, and in many cases, in long boilers, were only one-third or one-fourth the strength of the boiler. This anomalous condition of boiler construction will account for the numerous accidents that have occurred. It has now been remedied; and by a very simple and inexpensive process the flues may be strengthened to almost any degree of tenacity, by the simple introduction or attachment of T iron hoops at certain distances in the length of the flues.

"From these experiments I found that the resistance of flues or tubes varies in the inverse ratio of their diameters; inversely as the lengths, and directly as a power of the thickness. Or it may be stated that the strengths decrease in the ratio of the increase of the diameters and the lengths, and increase nearly as the square of the thickness of the plates. The general formula for calculating the strength of wrought iron tubes is, where

P = collapsing pressure in lbs.
K = thickness of plates in inches.
L = length of tube in feet.
D = diameter in inches, we have

$$P = 806{,}300 \frac{K^{2\cdot19}}{L\,D};$$

or it may be calculated by logarithms, in which case it may be written,

$$\text{Log } P = 1\cdot5265 + 2\cdot19 \log 100\,K - \log (L\,D).$$

"To illustrate this remarkable law; if we take three flues perfectly similar in every respect, one 10, one 20, and the other 30 feet long, we shall find the first twice the strength of the second, and three times the strength of the third.

"It will not be necessary to pursue this part of the subject farther, except only to direct attention to the following tables, which have been constructed from the experiments bearing directly upon the elastic force of steam, internally as relates to tension, and externally as relates to the collapse of the flues.

"TABLE III.

"Table showing the Bursting and safe Working Pressure of Boilers, as deduced from Experiment with a strain of 34,000 lbs. on the Square Inch as the ultimate Strength of Riveted Joints.

Diameters of Boilers.		Working Pressure for ⅜ Inch Plates.	Bursting Pressure for ⅜ Inch Plates.	Working Pressure for ½ Inch Plates.	Bursting Pressure for ½ Inch Plates.
Ft.	In.	Lbs.	Lbs.	Lbs.	Lbs.
3	0	118	708¼	157¼	944¼
3	3	109	653¾	145¼	871¾
3	6	101	607	134¾	809½
3	9	94½	566½	125¾	755½
4	0	88½	531	118	708¼
4	3	83¼	500	111	666½
4	6	78¾	472	104¾	629½
4	9	74½	447¼	99¼	596¼
5	0	70¾	425	94¼	566½
5	3	67¼	404¾	89¾	539½
5	6	64¾	386¼	85¾	515
5	9	61½	369½	82	492¾
6	0	59	354	78¾	472
6	3	56½	340	75½	453¼
6	6	54¼	326¾	72½	435¾
6	9	52¼	314¾	69¾	419½
7	0	50½	303½	67¼	404½
7	3	48¾	293	65	396¾
7	6	47	283¼	62¾	377½
7	9	45½	274	60¾	365½
8	0	44	265¾	59	354
8	3	42¾	257½	57	343¼
8	6	41½	250	55½	333¼

"Rule for $\frac{3}{8}$th inch plates.—Divide 4250 by the diameter of the boiler in inches; the quotient is the working pressure, being one-sixth the strength of the joints.

"Rule for ½ inch plates.—Divide 5666·6 by the diameter of the boiler in inches, and the quotient will be the greatest pressure that the boiler should work at when new; that is, one-sixth the actual strength of the punched iron.

"The above table may be considered practically safe for the construction of boilers of good iron, to be worked at the pressure indicated in the second column; and the following table of equal strengths of cylindrical flues may also be relied upon for a collapsing pressure of 450 lbs. per square inch.

"TABLE IV.

"Table of Equal Strengths in the Cylindrical Flues of Boilers, from 1 to 4 feet in diameter, and from 10 to 30 feet in length, showing the requisite Thickness of Metal for a Collapsing Pressure of 450 lbs. per square inch.

Diameter of Flue in Inches.	Collapsing Pressure of Flue in lbs. per Square Inch.	Thickness of Plates in Parts of an Inch.		
		For a 10 Feet Flue.	For a 20 Feet Flue.	For a 30 Feet Flue.
12	450	·291	·399	·480
18	450	·350	·480	·578
24	450	·399	·548	·659
30	450	·442	·607	·730
36	450	·480	·659	·794
42	450	·516	·707	·851
48	450	·548	·752	·905"

It is noticed that Mr. Fairbairn states: "On submitting a series of cylindrical and elliptical tubes to external pressure, they were weak, and in many cases in long boilers were only one-third or one-fourth the strength of the boiler." Now, on reflection, there need not

be any surprise or wonder at the results produced, natural laws being the cause for the effect attained. It has been stated in page 128 "that the elder, the reed, and the wheat stalk," amongst other illustrative members of the vegetable kingdom, are emblems of strength.

It must be noticed that, not only is the tubular form preserved, but extra means to cause strength are interspersed at requisite distances throughout the length of the stalk. And there Nature shows her beautiful simplicity of construction, which to be appreciated must be clearly understood.

These interspersions alluded to, are proportionately arranged to the length and diameter of the stalk, forming the same acquisition as T irons in a boiler tube; and not only therefore is this effective naturally, but mechanically also.

Mr. Fairbairn doubtless acknowledges this when he states: "By a very simple and inexpensive process the flues may be strengthened to any degree of tenacity by the simple introduction or attachment of T iron hoops, at certain distance in the length of the flues."

The positions of the details within the shell, as arranged in the example now under notice, are for cylindrical fire boxes, semi-circular combustion chamber, and also the smoke box if desirable. The crowns of the latter compartments must of course be sufficiently stayed either by cross or perpendicular supports, while the ends of the shell require stays to connect them; and the correct proportion of the stays must bear a strict relation to the pressure of the steam and the area exposed.

In some instances "gussets" are introduced —forming connections with the ends of the shell and the fire boxes and smoke box; and it is due to this connection that gussets are objectionable, because the resistance is divided at right angles, and therefore not direct.

Now, with longitudinal stays, the plates are connected directly, and thus the strain and resistance are equal in position.

The action of the flame next demands attention. The crowns and sides of the fire boxes are operated on as with other arrangements. The combustion chamber is the least effective in this instance, owing of course to the proportionate surface exposed; but its crown is operated on equal to that of the furnaces. The sides or ends are the least in relative value to the other surfaces, due to the sliding contact of the flame. The tube plate receives the flame at right angles to the line of progression, and the action, therefore, is the most effective. Much depends, however, on the velocity of the gases, and the proportionate area of the tubular passages to the grate surface as to its actual effect.

The effect of impact in this case is readily obvious; but the attainment of the same is the principal endeavour.

And the value of the production of direct action of the flame has been fully discussed in pages 124 to 126, therefore the gain in the present arrangement will be apparent.

The tubes receive the flame with the usual action, but the portions nearest the combustion chamber are the most affected.

The effect of the flame's action on the surfaces exposed must next be described. The crowns of the fire boxes and combustion chamber are equal in value for evaporation. The sides of the furnaces—above the grate—are of the usual ratio.

The ends or sides of the combustion chamber

are more effective, due to their being in a line with the sides of the respective furnaces alluded to, than with prior arrangements.

The effect of the flame on the tube plate, proportionately to the direct action of the flame, is also proportionately increased.

The late Mr. Williams made some practical experiments on the efficiency of the effective surface of tube plates for evaporation; and in pages 25, 26, and 27, allusions and quotations have been made on this subject.

Mr. Williams's experiments—as shown in his work—were made with tube plates situated at right angles with the line of the flame's progression, and in relation, he states:—

"The first experiment made was to ascertain the full evaporative value of the *tubes*, apart from that of the furnace or fire box. In a furnace 30 inches by 18 inches wide, 420 lbs. of coal were used in three hours, and 1,970 lbs. of water evaporated from 212°. The temperature of the escaping products in the chimney, as indicated by Gauntlett's pyrometer, was 1,060°, the water evaporated by each pound of coal being 4·69 lbs.

"From the high temperature in the chimney —106,0°—it was evident that much of the heat that should have been appropriated in generating steam had passed away as *waste*. Here, then, was a favourable opportunity for ascertaining if any of that heat could be utilized.

"For this purpose a small supplemental boiler was annexed, similar in every respect to the larger one, with the exception of having tubes of but 2 feet long instead of 6 feet. This reduction in the length of the tubes was made for the purpose of obviously throwing on its face-plate the work of evaporation. Between the two boilers was left an interval of 18 inches, as a heat box, to allow the heated currents to collect and strike the second face-plate, as it did in the first instance.

"The result of this arrangement, and the presence of a *second* face-plate was, that although the force of the draught was reduced by reason of the loss of the appropriated heat, and but 364 lbs. of coal used (against 420 lbs. in the previous experiment), the evaporation was *increased* from 1,970 lbs. to 2,080 lbs. as follows:—

From the 6-feet tube boiler ...	1,820 lbs.
From the 2-feet tube boiler ...	260 „
	2,080 „

"Here there was a gain of 110 lbs., or 5 per cent. more water evaporated, with a reduction of 56 lbs. in the coal used, or 14 per cent. The equivalent of this was shown in the reduction of the temperature in the chimney from 1,060° to 760°, and the evaporative value of each pound of coal increased from 4·69 to 5·69 lbs.

"Now, the strong ebullition which continuously rose behind the face-plate of the supplemental boiler showed that it was there the increased evaporative effect was produced.

"The third experiment was intended to test the heat-transmitting power of a *third* face-plate, in the expectation of utilizing still more of the escaping heat.

"For this purpose, in place of the 6-feet tube boiler, with its one face-plate, a boiler was made having *two face-plates*, as if the 6-feet tubes had been divided into two portions, each having tubes of but 3 feet long, with an intermediate space of 18 inches, to act as a heat

ber, thereby allowing the heated pro-
, from the first series of twenty-five tubes,
ite and strike the second face-plate as be-
To this was annexed (as in the preceding
·iment) the supplemental small boiler,
giving the effect of a *third* face-plate.
[he result was proportionately satisfac-
The combustion of 392 lbs. of coal
ıced a still further evaporation, as
vs :—

ı the two-face plate boiler, evaporated	2,200 lbs.
ı the small supplemental boiler	240 ,,
	2,440 „

Still more of the escaping heat was
ed; the temperature in the chimney
; further reduced from 760° to 635°, and
vaporative value of each pound of coal
ased from 5·69 to 622 lbs."
om these results it seems apparent that the
late is the principal agent for evaporation.
is stated in page 28, that—in allusion to
present subject—the gain or increased
orative power, with intermediate combus-
chambers, was not "due to the face or
plates," but rather to the "time" allowed
mbustion. This fact is further rendered
us on remembering that the congrega-
of the gases produced rapid and almost
ct combustion. Apart from this, the flame
n the tubes is *checked* in its progress by
'expansion" and "reformation" of the
nes in the respective chambers. The
es," therefore, *not* the tube plates, have
generally considered the chief agents
he penetration of the caloric, but Mr.
iams (all honour to his name as an able authority!) endeavoured to explain away this by stating :—

"There is a fact which, although hitherto unobserved, here merits special notice, as it exhibits a remarkable contrast between the action of the gaseous current passing *through* the tubes in the one case, and *against* the face-plate in the other; namely, that in proportion as the chimney draught is *increased*, so is the heat transmitting power of the face plate surface, whereas that of the tube surface is *diminished.*

"This will readily be understood, when we consider that in the tubes the heated current passes, with extraordinary rapidity, *along the plane of their internal surface;* whereas, in the face plate, it strikes with *a direct force* at right angles to that surface. The same may be said of flue boilers, and which are often made of no less than thirty feet in length. In these the current of heated products passes along the plane of their interior surface, manifestly, as in the case of tubes, giving out but little heat, *laterally*, to the course of that current.

"We here at once recognise the cause of the great steam generative power of the *single face-plate* in each locomotive, namely, the almost electric rapidity of the draught in the chimney, producing a corresponding increase in the force with which the heated current strikes that plate."

Now the natural direction of caloric is to ascend, and when the flame was passing through the tubes the penetration of the heat was therefore the most available.

The action of the flame on the tube plate could not have been with a *direct force*, in the

strict meaning of the term, but rather as stated in page 113, that "minute curved hollows are formed and destroyed in succession with sufficient velocity to ensure the *actual contact* of the flame on the surface between the tubes." It is, therefore, understood that "actual contact" can only be attained but by "velocity" of the flame; and correct proportion of the area of the tubular passage to the grate surface. The former acquisition is readily attained with the "direct tubular" arrangement, but with other arrangements the value of the tube plate becomes duly lowered.

The results attained by Mr. Williams would doubtless have been much inferior had a return tubular arrangement been adopted. Not only would the draught have been lessened, but the impact also reduced.

Simplicity of construction must also be considered, and access for repair especially demands attention with marine boilers.

What may seem an advantage for land purposes will be adverse for steam vessels, and more particularly so in the event of an accident at sea.

The values of the heating surfaces of the arrangement now under direct notice, from the conclusive evidence presented, are considered to be in consecutive ratio, as follows:—

TUBES.		
Total Surface	... ...	2·976
FIRE BOX.		
Crown ...	Total Surface ...	1·000
Sides ...	Above grate ...	·500
COMBUSTION CHAMBER.		
Crown ...	Total Surface ...	1·000
Ends or sides	" ...	·400
Tube plate	Effective surface	·900

CYLINDRICAL-SHELL AND FIRE-BOX. RETURN-TUBULAR-ARRANGEMENT.

The advantages of a tube-like form of shell and fire box for boilers have been freely commented on in pages 127 and 128. The arrangement, therein expressly alluded to, was the direct tubular kind; but the present purpose is to prove the practicability of a return action for the flame, combined with the greatest strength of structure and access for repair.

The illustrated sections, Fig. 20—page 137—represent a type of low boiler especially adapted for vessels of light draught and narrow beam; while space is economised as much as practice admits, and strength is ensured by the form adopted.

The shell and fire box are tubes of proportionate diameters pertaining to the respective requirements. The fire grate is supported in the ordinary manner, with the usual brick bridge at the back end. The combustion chamber is semicircular from the roof, the latter being flat, while in some instances a curved form is introduced. The tubes almost surround the fire box, as seen in the transverse section. The smoke box is at the front end of the shell, forming a projection around the fire door, and the uptake ascends centrally of the boiler front.

The number of stays requisite, it is seen, is very much less than with prior examples illustrated. This is not only a reduction of material and expense of construction, but also access for inspection and repair is rendered certain.

The crown of the combustion chamber being flat in this instance, longitudinal support of the same is requisite; and it is deemed preferable

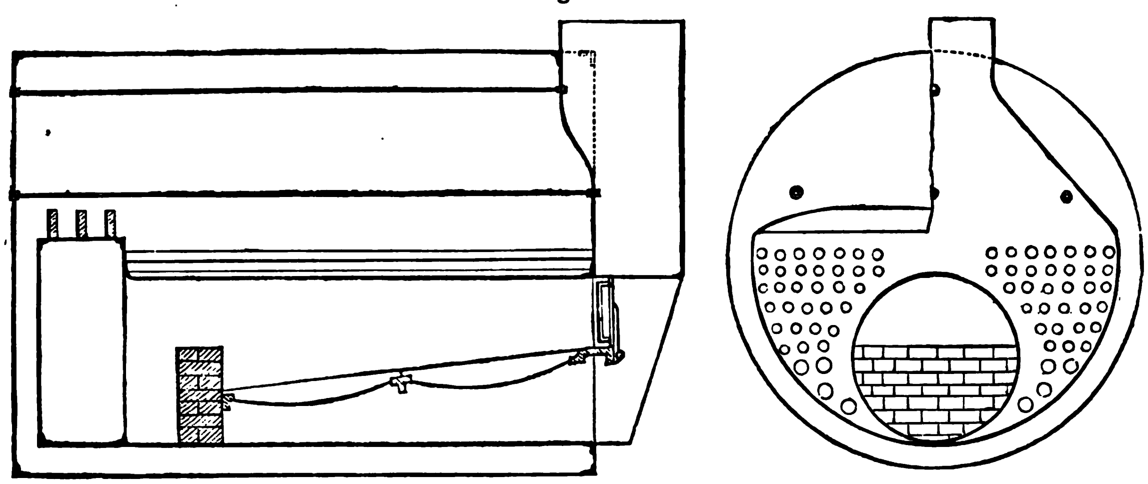
in this example to attain the same by cross bars, rather than vertical stays connected to the shell.

Now, it may be considered by some authorities, that any connections with the shell as a support for any of the parts or surfaces of the internal portions, is a truthful illustration of the proverb that "unity is strength." With boilers, however, the rule should be—independent strength due to the form of the internal structure without connection with the shell if possible.

The back of the combustion chamber is the attainable with a proportionate thickness of plate, independently of staying. Now, with the flat shell, this rule is far from absolute; because if in that case an increase of pressure must be resisted, it can only be maintained by securing the stays at a pitch, proportionately to the surfaces exposed to the pressure of the steam—the requisite increase for the thickness of the plates, laps of plates, and riveting, being of course also duly considered.

Having thus explained further the gain there is with cylindrical boilers, a considera-

Fig. 20.



RETURN TUBULAR CYLINDRICAL MARINE LOW BOILER.

only internal surface, with the position of the details illustrated, that requires stays for intermediate resistance to the common pressure. These stays, therefore, are not only adequate for that purpose, but they also form a support for the end of the tube or fire box.

This simplicity of construction, combining strength, with the form of shell in question, is obviously not produced with any other arrangement of previous explanation. Also, it is certain that, with the tubular form of shell and fire box, an increase of strength is directly

tion of the objections presented with the same will be of value. The principal objection, as previously stated, is the waste of the spaces between the shells, for marine boilers, which of course is more condemnable with a number of boilers than when one or two are used; because without an intermediate space, available for coal bunkers, the waste of cubical capacity is increased, but on the contrary when the shells are arranged in continuous contact with each other. Now, to further exemplify, presume a boiler is required of 100

horse power, nominal, and it is determined that four grates shall be introduced. The total amount of grate surface is taken at 78 square feet, then $\frac{78}{4} = 19{\cdot}50$; length of grate to be 6 feet 6 inches, and the width of 3 feet, which is the diameter of each fire box—then the diameter of the shell will be about 13 feet 6 inches, which is now no novelty of size combined with strength.

The fire boxes being situated parallel, both transversely and longitudinally, with the outline of the shell, the tubes can be disposed in one unbroken cluster, above and between the fire boxes, to attain a return action for the flame. The length of the shell will be about 9 feet 6 inches, similar as for an ordinary flat high boiler of the same power.

The actual cubical space occupied by the cylindrical shell is, therefore, about 1360 feet. The cubical space occupied by a flat shell boiler of the same power is about 1515 cubic feet. It may now be said that the advantage, as to the cubical space occupied, according to the figures quoted, is with the cylindrical shell. It must not be forgotten, however, that with the flat form, no space is wasted. With the former example it must be remembered that the space below the centre line of the shell, longitudinally of the same, is of no practical utility. When the peripheries of the shell are in contact, as before stated, this evil increases in due proportion.

Now, to prove the latter remark to be a fact, presume four boilers of each form are required, each boiler 100 nominal horse-power: the shells of the flat exterior occupies 6060 cubic feet, actually *without any waste:* the cylindrical form occupying 5440 cubic feet *with waste* spaces between the shells, above and below the centre line. The spaces longitudinally of the shells, outside of the same, also are of no utility. The total, amounting to 1152 cubic feet for *waste* space, added to that utilized = 6592 cubic feet, thus proving that the boiler with the flat sides, &c., occupies 532 cubic feet less than the cylindrical form.

The latter also requires a height of 13 feet 6 inches, and the length at the base is 9 feet 6 inches. Now with the flat shell the length at the base can be reduced to 6 feet, or even less, the back forming an angle convenient to the rise of the ship's side. The actual height also required for the flat shell being only 11 feet from the base to the roof of the boiler.

The remainder of the requirements attained, in comparison with the objections produced, being common with other arrangements, further demonstration is unnecessary.

It is noticed in the illustration, Fig. 20—page 137—that a few tubes are omitted directly over the centre of the crown of the fire box. This omission is purposely, to attain the means of removing any scale that may accumulate on the surface in question. Doors, suitably placed, are arranged at the front and top of the shell for inspection and repair.

The larger tubes, that are secured at the base of the combustion chamber, are no novelty, being rather a recognition of a tried acquisition; the same being to prevent choking by the soot or condensed smoke.

The form of the shell and fire box: arrangement of the detail: and the defects and advantages common to the same, having been

duly commented on, attention must next be given to the action of the flame in the boiler.

The flame proceeding from the grate operates on the surface above the same, at an angle rendered higher or lower by the amount of draught admitted. The surface next in contact is the back and crown of the combustion chamber; and the portion of the former in front of the bridge receives the flame similar to that alluded to in page 111; the arrangement referred to—Fig. 16—being the same in principle as that now under direct notice: also it is there stated that "the flame, on entering the combustion chamber, is naturally divided in the form of curves, due to the position of the tubes."

The semi-circular portion of the combustion chamber will be mostly operated on above the level of the bridge, that below being the least in actual contact; and the crown will, of course, receive the greatest impact of the flame.

Next to be noticed is the tube plate; and the prior allusions as to the action of the flame on this surface, due to its locality, are also common to the present arrangement.

The position of the tubes is doubtless strictly conformable with natural laws; because all volumes, on entering the combustion chamber, tend to ascend; and the top rows of the tubes of all boilers receive the greater portion of the purer flame; and it is seen that, in the present example, this is especially provided for, not, perhaps, however as much intentionally for that purpose as advertently to the arrangement.

The action of the flame within the tubes is, of course, due to the draught produced; and it is certain that the surface in contact with the flame, viewed longitudinally, presents an undulated curve, three-fourths of the tubes' length; while its action in the remainder forms an angular line to the smoke box in this manner.

Transverse sections of the surfaces acted on present the following: at the tube plate, a complete circle; at the centre of the first undulation, a semi-circle; for a given length beyond the rise of the curve, a complete circle; centrally of the angular line formed, a second semi-circle. It may be added that this theory is congenial with that alluded to in pages 89 and 90, and is indeed common to all the prior and future allusions bearing reference to tubes horizontally situated.

The effect of the flame determines the values of the heating surfaces with all arrangements, and a brief comment in the present instance will be requisite.

The under flame line of the volume, as it passes the bridge is, doubtless with an ordinary draught, almost straight, connecting the inside edge of the bridge and the base of the combustion chamber. The rise or fall above or below the line in question is of course due to the air admitted above or below the grate; and the amount of surface affected, relative to the fire box, may be considered one half of the total.

The back of the combustion chamber is, doubtless, more or less fully exposed to the effect of the proceed from the grate; and the impact is due to the velocity of the volume. The crown is of the same value as the surface produced in the fire box; also the tube plate is of proportionate ratio, common to the relative position; while the least in value is the sides, the surface of which, actually effective, may be taken as three-fourths of the total.

The total surface of the tubes is entirely noticed, being of the same value as those of precedent notice.

From those conclusions, the values of the respective surfaces are readily known to be as follows:—

TUBES.

Total surface ...	... ...	2·976

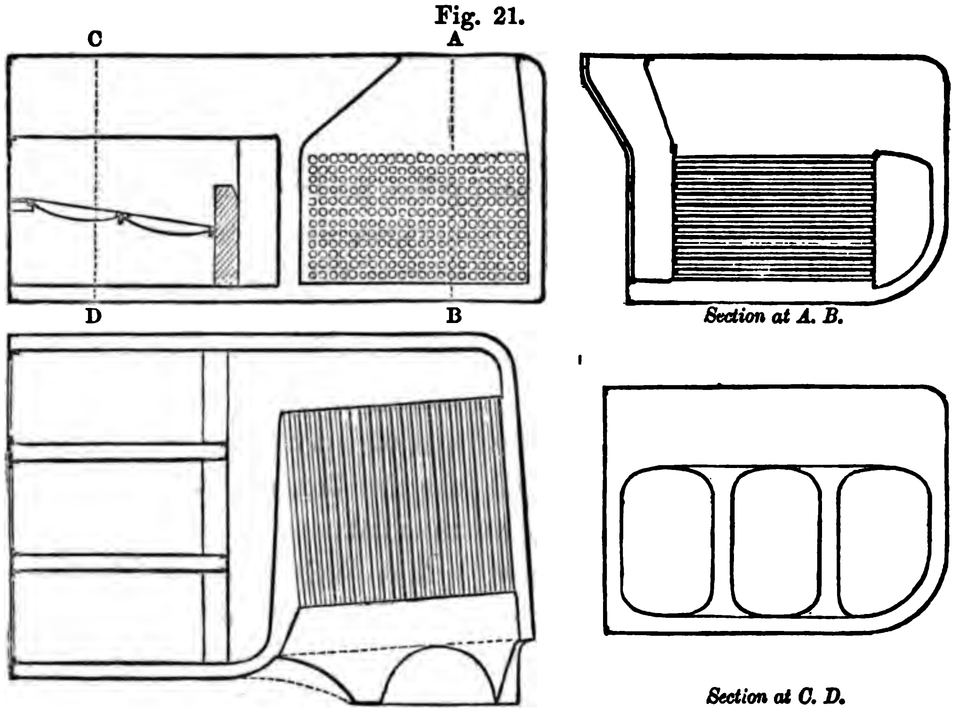
FIRE BOX.

Half surface ...	... ...	1·000

COMBUSTION CHAMBER.

Crown ...	Total surface ...	1·000
Tube plate ...	Effective surface ...	·875
Back plate ...	Total surface ...	·700
Side ...	Three-fourths' surface	·300

Fig. 21.



MESSRS. RENNIE'S ANGULAR TUBE MARINE LOW BOILER.

The respective ratios are thus rendered obvious, without further comment being necessary.

ANGULAR TUBE ARRANGEMENT.

The arrangement of "low boiler," now to be described, Fig. 21, is nearly similar to that alluded to in page 96, and illustrated by Fig. 14. To render the comparison as to the difference obvious, the portions only requiring that especial notice will be alluded to.

Now, with reference to the sectional plan in Fig. 14, the side of the shell opposite the tubes is curved, in the present under notice it is straight.

The connection of the side and back end in the prior example is at right angles; in the present a curve is deemed preferable. The curve joining the side of the inner fire box, and the divisional plate, in the present arrangement, is of a larger radius than in the former. The tubes in this instance are angular with the centre line of the hull; in that previously alluded to, a perfect right angle is retained. The smoke box also presents a different aspect, due to the length of the tubes.

In the relative sectional elevations, or as at A B, not much variation in form presents itself for comment. The base and back portions, in the present example, are connected by a curve; whereas, an angle is preserved in the former example. The termination of the uptake projects from the smoke box doors, as seen at present, rather than a vertical passage as with that previously.

The remaining portions are duplicates of that depicted by Fig. 14, as far as the internal arrangement; but the fittings and detail are omitted as superfluous for the present purpose; therefore from the various sections given, a correct conclusion can be attained without further description.

The action and effect of the flame, the value of the heating surfaces, and the apparent gain throughout, is much the same, in the present arrangement, as with that alluded to as a comparative example.

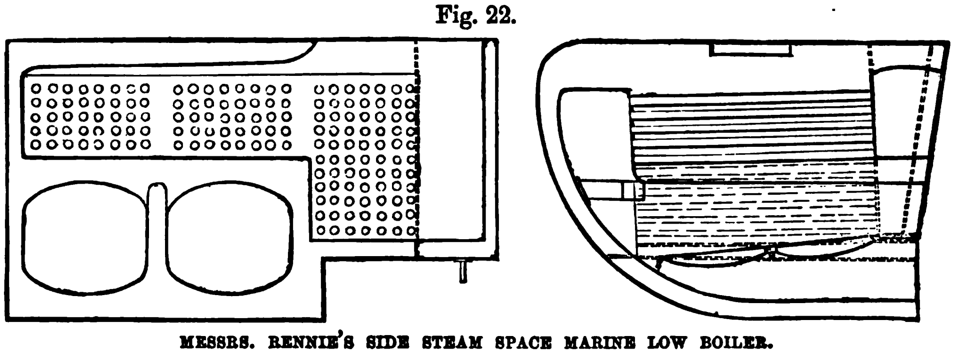
RETURN TUBULAR—SIDE STEAM SPACE—ARRANGEMENT.

It is well known that compactness of arrangement is one of the main considerations relative to the low boiler, and that ample steam room is often inadmissible, therefore a peculiar means for the largest room possible is often resorted to. The illustration below, Fig. 22, is a fair example as to what can be done in the way of distributing the portions for a marine boiler; because, not only is the steam space laid on its side, but the tubes also are peculiarly disposed.

It will be remembered that with all the prior examples alluded to, the tubes are either above or on a level with the fire box. In the present instance it is seen that an amalgamation of those situations is preferred.

The form of the shell also presents a novel aspect, both in side and end elevations. The "steam space," with ordinary arrangements, is directly over the water, but in this present example it is at the side—not common to the fluid compartment, but a separate receptacle is provided—the shell containing the entire portions.

Fig. 22.



MESSRS. RENNIE'S SIDE STEAM SPACE MARINE LOW BOILER.

In order to render the present arrangement obvious, the following description is given. The fire boxes are arranged for stoking athwart ship. The grates are of the ordinary kind, with the absence of the brick bridges. The transverse section of the combustion chamber forms a striking contrast with those of previous allusion. The general practice is to make the width less at the crown than at the bottom of the tube plate; with the present example it is seen that

the greater breadth is at the top of the chamber, —this, doubtless, is as much to preserve access for repair as notion of design.

The arrangement of the tubes can be better understood by referring to the sectional elevation. It is seen that above each fire box are arranged a given number of tubes in the ordinary manner. To reduce the total height of the shell, the remainder of the tubes are arranged at the side of the inner fire box; the top row of the entire set being in a straight line, and thus one level is preserved. The level of the bottom row of the side cluster is at about that of the fire grates.

The smoke box partakes also of the novelty of design common to the remaining portions, due, of course, to the arrangement of the tubes.

Apart from the tubes being so disposed to reduce the height of the shell, the space between the roof and the water line is also contracted to a minimum, being about seven or eight inches. Now it is evident that this height is inadequate for the steam room requisite.

The surmounting of difficulties is always a triumph in all grades of pursuit, and, often, none are more worthy of congratulation than the designers of marine engines and boilers. It often occurs, however, that the gain certain in one locality causes an imperfection in another; it is therefore obvious that the acquisition of desiderata common to the whole arrangement is no ordinary attainment. How far these remarks apply to the example of low boiler under notice needs no comment for the present purpose. It is sufficient to recognize facts without personal opinions.

The water line in the boiler now under direct notice is stated to be so near the roof, that a separate steam space is requisite, which is attained by a compartment being formed at the end of the shell, the communication for the traverse of the steam being at the roof. It is seen also that the steam space has a casing surrounding it, separate from the water compartment in the boiler. At the base line a pipe is secured for the purpose of discharging any water or condensed steam that may accumulate within the casing in question.

The arrangement under notice is compact in the full meaning of the word, relative to the vertical positions of the details especially. It must be noticed, however, that the correct total cubical capacity, proportionately to the area of the grate surface, is unalterable with all boilers. The space that is generally above the tubes with other arrangements, is at the side or end of the shell in this example. Room in the hull is therefore required, longitudinally rather than in any other direction.

Having briefly commented on the arrangement in question, allusion will next be given to the action of the flame. In the present example, two lines of progression are in operation at the same time: the one at right angles vertically, and the other horizontally with the furnaces. These variations are, of course, due to the position of the tubes.

The flame, on entering the combustion chamber, has to divide itself into two volumes, one pertaining to the tubes directly over the fire boxes, and the other to the tubes at the side. Now the situation of the uptake—as before noticed—to a great extent governs the action of the flame.

With the present arrangement, the fire boxes

are similarly placed, in relation to the uptake, as with high boilers, the action of the flame is therefore much the same in each instance. The exception in the present case refers especially to the tubes below the level of the crowns of the fire boxes. It is understood that, to render these tubes effective, a given amount of heated gases must enter them. Now the situation of the uptake in the present example prompts the idea that the tubes near the base line of the shell cannot receive much of the proceed from the grate, other than smoke and soot. The top rows of tubes always receive the maximum volume, due to their situation; and the action of the flame, when in the uptake, is common to other arrangements, which have been already dwelt on.

With reference to the effect produced, a brief allusion will not be out of place. The proceed from the grate affects the crowns and side surfaces of the fire boxes similar to that with other arrangements.

The back of the combustion chamber, forming a continuous curve with the bottom of the fire box from the grate, the effect of the flame on that surface is thus rendered of higher value than with a vertical action.

The crown surface is equally effective as that of the fire boxes. The sides or ends of the combustion chamber are of increased value as agents of evaporation proportionately to other surfaces.

The effective area of the tube plate is of proportionate value, relative to the position of the tubes. The total length of the plate in question multiplied by the width above the fire boxes, and the areas of the respective tubes subtracted = the surface worthy of most consideration for the present purpose. The remaining area exposed is of the least value, and considered to be one half of the former. The cluster of tubes above the fire boxes is of the ordinary value, while those at the side are of the same ratio as the tube plate at that locality. Having thus far rendered certain the values of the surfaces in question, the following classification of ordinary ratios is deduced :—

TUBES.		
Cluster above the fire boxes ...	...	2·976
Cluster at the side of the fire box	...	1·488
FIRE BOX.		
Crown...	...Total surface... ...	1·000
Sides ...	...Above grate... ...	·500
COMBUSTION CHAMBER.		
Crown...	...Total surface... ...	1·000
Tube plate	Surface above fire boxes...	·875
„	Surface at side of fire boxes	·437
Back plate	Surface in front of furnaces	·720
Ends ...	Surface above grates ...	·250

DIRECT RETURN TUBULAR ARRANGEMENT.

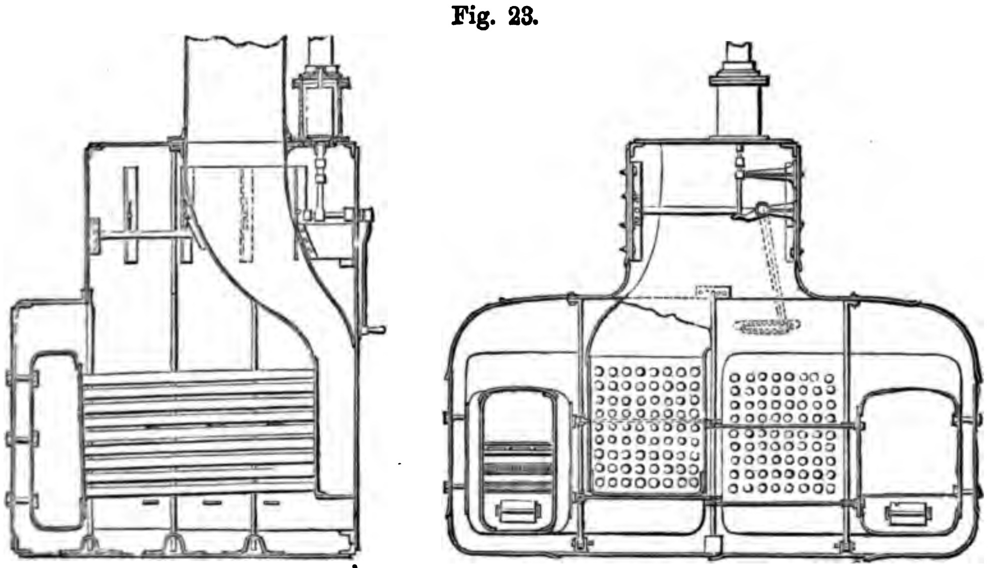
To render the arrangement of any portion of mechanism or structure conceivable by illustration, without mistake as to its purport, three views at least are requisite. It is for this purpose the sectional elevations, Fig. 23—page 144—are introduced. In page 115, a sectional plan is given, in common with the principle of the arrangement now under notice. In that case two fire boxes are on each side of the tubes, while only one is shown at present. The boilers in each instance are nearly similar in design, but of different power and capacity. The roof of the boiler in the prior example is entirely flat, whereas in this instance a given portion of the top is raised to form a dome,

especially for the reception of the steam. The tubes also are raised higher from the base line in this case than in that of previous allusion, doubtless for the purpose of forming a recess in the combustion chamber, below the lower row of tubes.

Thus far the exceptions are disposed of, and the next step is the description of the arrangement in question. On noticing the transverse sectional elevation, it will be seen that the top row of tubes is above the crown of fire boxes; for the ordinary twofold purpose of preventing adhesion and permitting an escape of steam at a lower pressure than required to lift the valve naturally, due to the load on the same.

The other sectional elevation represents the combustion chamber, smoke box, and uptake in section. The tubes connecting the two former, it is noticed, are at an incline, for the purpose of accelerating the draught. The recess, already alluded to, below the bottom row of tubes, is for the reception of soot, &c., that

Fig. 23.



MESSRS. WATT'S RETURN TUBULAR MARINE LOW BOILER.

also the crown of the combustion chamber is similar. The sections given of the fire boxes are centrally of the length and at the front end, to show the width in each instance. The doors seen under the grate, at the back ends, are for the admission of air beyond the bridge, a suitable framework being fitted in the fire box. The stays are situated according to their several requirements; disconnection being attained by the removal of the pins or cotters. The gear shown in the dome, and that in dotted lines below it, relate to the safety valve, being passes from the furnace during imperfect combustion.

The smoke box is in two compartments, from the base to the upper row of tubes, where a connection is formed, and the uptake thus rendered common to both sets of tubes. This is a contrast with Fig. 17—page 115—where is shown, in sectional plan, that each smoke box is separate, in front of the tubes.

In the view now under direct notice, the uptake is curved, the steam thereby surrounding it, which renders partial super-heating

attainable without extra detail or expense. The safety valve and gear shown in front of the uptake is the ordinary kind, presenting no novelty for description or comment.

The stays are arranged between the tubes and beyond the same at given distances, relative to their sectional areas and the pressure to be resisted.

This description of the arrangement being sufficient for the present purpose, a brief comment on the action and effect of the flame, and values of the heating surfaces, will next be given.

On referring to the plan of the boiler in page 115, it can be seen that the combustion chamber is at right angles to the grates and tubes, which also is the position of those details in the present instance. The flame, on rising from the fuel, acts against the crown and sides of the fire-boxes. Directly after passing the bridge, it forms a curve at right angles to its former line of progression.

Now it is due to this curve, as before stated, that the portion of the fire box beyond the bridge connecting the back of the combustion chamber—is the least affected by the flame. The ends of the combustion chamber are also similarly operated on.

From the cluster of the tubes being raised in relation to the crown of the fire boxes, doubtless the lower rows receive a purer flame than when situated near the base line of the combustion chamber.

The values of the heating surfaces may next be concluded. The fire box is of the same ratio as that of prior notice. The combustion chamber, however, is not of equal value, proportionate to its entire surface. This conclusion is arrived at from the fact that the flame, during exit, inclines towards the uptake. It is obvious, therefore, that the volume on passing the bridge acts only on the crown, tube plate, and that portion of the back above the level of the lower row of tubes. The ends of the combustion chamber are, of course, the least in contact with the proceed from the grate, three-fourths of the total surface being only worthy of attention.

The effect of the flame on the crown, back, and tubes plates is much the same as with similar dispositions of detail, the values, therefore, are alike; remembering the proportionate surfaces in contact.

The tubes, of course, admit the passage of the flame through them with a given effect, due to their position; and the flame-value is from that cause readily deduced.

To render the matter under question concise as well as practical, the following synopsis of the proportionate ratios is added, uniform with those previously given.

TUBES.

Total surface...	... ...	2·976

FIRE BOX.

Crown ...	Total surface ...	1·000
Sides ...	Above grate ...	·500

COMBUSTION CHAMBER.

Crown ...	Total surface ...	1·000
Tube plate	Effective surface surrounding the tubes, and surface above...	·875
Back plate	Surface above the level of the bottom row of tubes	·700
Ends ...	Three-fourths' surface	·250

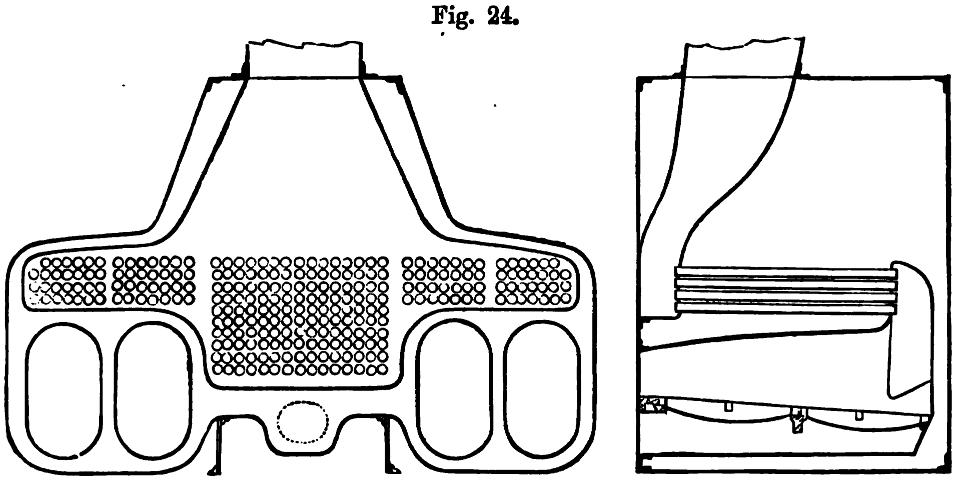
SEMI-HIGH AND LOW RETURN-TUBULAR ARRANGEMENT.

When solving any problem, whether pertaining to mathematics or mechanics, the observation of natural laws tends not only to assist investigation, but to lead to a truthful result. Now, no member of any profession is under a greater obligation to nature than the Engineer. She teaches him that, in all her examples of production, whether belonging to the animal, vegetable, or mineral kingdoms, a lesson is ever before him worthy of research. Mankind, the birds and beasts, all liev under one law. The trees, flowers, and herbage all partake of the same means of existence. Geologists also agree on the question as to the universal law which governs the formation of the strata of the earth.

The marine boiler contains two elements—fire and water—and is itself a mineral property. It is from this cause that the Engineer, when dealing with the present subject in particular, should appeal to Nature for assistance. Not only is this a certain means of acquiring a truthful result, but it is actually the mode adopted by every function of each element. There is throughout the course of *natural* formations a *just recognition* of the *existence* and *requisition* of every portion in creation. The application of these remarks to the preliminary conclusions, when designing marine boilers, refers especially to the natural flow of gases and flame; consideration, after which, should be given to the arrangement of the tubes.

In all the prior examples of low boilers it is noticeable that the level of the lower row of tubes is at or below that of the fire grate. With reference to high boilers, it has been stated, in page 88, that—"the upper portion of the tube plate is undoubtedly mostly operated on by the gases of the highest temperature, and the bottom the lowest. Practical evidence of this is seen with the lower rows of the tubes, where they are almost choked with soot." The cause for this is stated to be—"the smoke passes direct under the tube plate connection to the nearest exit; and the flame being of lesser density ascends to the central and upper tubes." Now, these facts being worthy of notice when designing low boilers, the example now under allusion is carefully

Fig. 24.



MESSRS. WATT'S RETURN-TUBULAR MARINE LOW-BOILER.

arranged to realize the requirement alluded to; and the sectional elevations, Fig. 24—page 146—will assist the appreciation of these prior and present observations.

Before describing the arrangement in question, a brief notice will be given, relative to the cause and effect of given arrangements of boilers, this especially referring to the recognition of circumstantial arrangements rather than correct positions of the details.

The principal object in view with designers of low boilers is a given amount of evaporation within the least height or depth of boiler space. The area requisite in plan is often the least considered, hence the cause for such widely distributed arrangements as some of those of prior illustration.

The observer who finds fault with a given position of detail, too often hastily condemns the same; expressing wonder perhaps, on his part, that such a production was ever mooted, to say nothing of its existence.

Now due consideration—of which experience alone can teach the value—should be given to *all* productions. There certainly must be one advantage, if not more, due to the particular positions of the details, that is worthy of notice. An arrangement of boiler may be adverse to the truthful gain by the strict observation of natural laws, as far as the action of the flame is concerned; while for war or stated purposes the requirement may be attained to a great extent. It is with this question that the present remarks deal, in as much that to appreciate any arrangement, the main cause for the same must be duly acknowledged.

The type of boiler now under especial comment partakes of two positions of details, as the title of this section purports. The transverse sectional elevation clearly shows the arrangement of the tubes relative to the fire grates. It is noticed, also, that the lower row of the former are above the level of the latter, which, it may be added, is correct in principle.

Four furnaces supply flame, in this case, to one cluster of tubes—no intervention being introduced—therefore the volume from the furnaces is common to one means of exit. The sectional plan—through the fire-boxes—of this arrangement is nearly as that depicted by Fig. 17—page 115—the exception being that there is no central water space in the present example. The fire boxes are alike disposed in each case, but the tubes in this instance are situated at the sides of and above the furnaces, to which distribution allusion has been made.

The longitudinal section illustrates the fire-box and uptake in section, the tubes being in profile above the former. The uptake is curved back from the boiler front to allow the steam to surround it; and by this means partial super-heating is maintained.

Next to be considered is the action and effect by the traverse of the flame, and the values of the heated surfaces operated on. The entire proceed from each grate flows in one compartment, in front of the tubes. The volume, as it enters the chamber in question, tends to escape in a direct line to the uptake. It is due to the situation of the latter that the lower rows of tubes are doubtless effective, as much as from their being above the level of the grate; also the total area of the tubes directly above the fire boxes are insufficient to receive the entire volumes from the respective sources; and thus the intermediate tubes are rendered effective.

The flames from the two inside furnaces are doubtless the first to enter the tubes between them. Now this is not only due to the position of the details, but also that gases of separate densities do not cross each other, as before stated.

The volumes from the outer fire boxes are permitted to maintain more perfect combustion, than those nearer the uptake, due of course to the relative distances determining the time of progression; and a recognition of this effect is given in page 116.

The position of the uptake in the present example being central, each grate, on each side of the tubes, is of relative effect. The tube plate from this cause is considered to be operated on in due proportion to the draught attainable; or the entire effective surface is recognized as of evaporative value. The surface interspersed amongst the upper rows of tubes is of course the greatest ratio, owing to the natural flow of the flame being to ascend.

To further explain the question under notice will be superfluous, it being obvious that prior allusion to a similar arrangement suffices for the present purpose.

The values of the heating surfaces in the present example are much alike, as those of previous consideration, in page 117; the exception being, in this case, that the tube plate may be of little higher value, due to the respective locality of the tubes.

MID-STOKING RETURN-TUBULAR ARRANGEMENT.

It sometimes occurs that the production of a correct title for a given arrangement of boiler or engine is far from an easy task; often more particularly difficult when some portion of the positions of the details is a duplication of two or more examples having had previous comment. It is with the view to clearly convey a correct meaning to the mind, apart from optical demonstration, that the given title to the present section is introduced.

The arrangements of boilers illustrated by Figs. 14, 18, and 21, on pages 96, 118, and 140, are nearly similar in principle to that now to be described, and illustrated by Fig. 25.

The arrangements of the tubes and grates being respectively alike in each instance, Figs. 14 and 21 referring to the former, and Fig. 18 to the latter.

In the present example it is noticed, in the sectional plan, that the fire boxes are on one side of the tubes, the latter having a return action. The longitudinal section shows the position of the tubes to be as the prior examples comparatively alluded to.

The transverse sections are concisely illustrated, consequently need no description more than an allusion to the action of the flame, to which attention will next be given.

To clearly define the truthful line of progression of the volumes emanating from the respective sources, a consideration of the arrangement is essential. Enough however has been said before, perhaps, on this latter subject to impart sufficient information for the present purpose, the main object under notice being a correct mode of judging of the relative values of the heating surfaces in proportion to other definitions.

On referring to the sectional plan it is seen that the grates are common to one combustion chamber, and the latter to one cluster of tubes; it is noticed also, on alluding to the

sectional elevation, that the uptake is in the centre of the tubes, transversely above the same. Now the line of progression and line of impact are subservient to this disposition of the respective portions.

Assume that the ignition of the fuel is alike in each furnace, the following description of the action of the flame will impart a correct idea as to the value of the surfaces in contact.

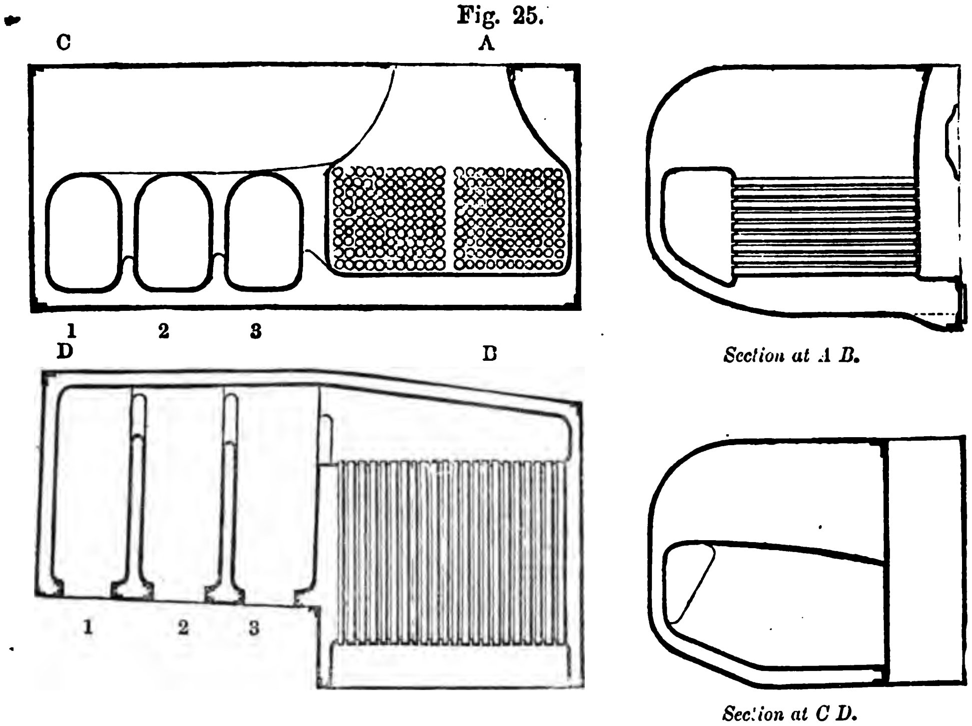
The volume from the furnace nearest the tubes—being nearest the uptake—is the first to enter the combustion chamber. Next to follow, is that from the middle grate; and, lastly, that from the furthest source. Now it is presumed that each product is in one compartment, at one side of which there is an allotted means of exit.

To assist memory and definition, the furnaces are numbered 1, 2, and 3, the first being furthest from the uptake.

The volume from No. 3 will be the first to enter the tubes, a given number of which will be occupied in conveying the same to the uptake. No. 2, next, sends forth its proceed, which enters the set of tubes beyond that of No. 3, or the central cluster. No. 1 furnace supplies the end stack of tubes, and from this it may be imagined that an almost independent flow is maintained.

Next, it will be well to consider the cause for this separation of the various volumes. In allusion to this subject in page 98, a brief comment was made: also, further in pages 121 and 122. On referring to the sectional plan of the arrangement on this page, it is seen that the outside grate, or No. 1, must be the most effective for evaporation, due to its locality. The volume from the inner, or No. 3 grate, being the nearest the uptake, the least time is permitted for combustion, although

Fig. 25.



MESSRS. WATT'S SIDE-RETURN-TUBULAR MARINE LOW BOILER.

the draught can be, of course, duly regulated. The centre fire box is the next in value, above No. 3 : the proceeds, from each source, are of relative densities, due to the time they remain in the combustion chamber. That from No. 1 grate, having the greatest traverse, is, of course, the lightest. The fact, therefore, that flames of separate densities do not cross each other, is the cause for the almost independent flow.

Having thus far treated of the progression of the flame, attention must next be given to the action of the same on the various surfaces. The contact of the volumes with the crowns of the fire boxes and combustion chamber is alike. The sides of each furnace above the grate, nearest the tubes, will be the most operated on, due to the position of the uptake. This fact is again another truthful illustration of the effect of natural laws. Also, clearly indicating the direct line of the flame's progression.

The action of the flame on the portion in front of the furnaces is conducive of greater impact than on that in front of the tubes. This effect is due to the line of progression being curvous—or, leaving the back plate—in front of the tubes, and sliding, combined with impact at the front of the bridges. The proceed from the furthest furnace—or No. 1—is most in contact with the back, the remaining proceed being less capable of acting on the same, due to the arrangement.

The ends of the chamber in question are the surfaces least operated on ; the reason for which has been freely treated in previous allusions.

The tube-plate now demands consideration. Three volumes of separate densities and powers are said to be in operation against this surface. To divide the same into relative values will be of no gain ; therefore it is better to acknowledge the total effective surface, and represent it by a given numeral.

The action of the flame within the tubes is the same as in those of prior allusion where the position of the detail is alike. There is not the least doubt that, with all arrangements, the upper rows of the tubes receive the purest flame, and the impact in the plate is also the greatest at that portion. This has before been alluded to, and in conclusion it may be added that the distribution of the tubes below the fire grates should be avoided, in all cases, if practicable.

For practical purposes, sufficient has been stated to render a truthful conclusion obvious as to the relative values of the heating surfaces now under direct notice.

	TUBES.		
Total surface	... ...	...	2·976
	FIRE BOX.		
Crown	... Total surface	...	1·000
Sides	... Above grate	...	·500
	COMBUSTION CHAMBER.		
Crown	... Total surface	...	1·000
Tube plate	Effective surface	...	·875
Back plate	Total surface	...	·700
Ends	,, ,,	...	·250

It can be noticed that these values are precisely as those in page 117, which refer to the arrangement illustrated by Fig. 17.

To make the foregoing conclusions of practical utility, the names of the makers and the ships fitted by them are given in the following table :—

No. of Fig.	Page.	Name of Ship.	Maker of Boiler.	No. of Low Boilers.	Nominal Horse Power, collectively.	Length of Shell.		Width of One Shell.		Height of Shell.		Total No. of Fire Grates.	Length of Grate.		Width of Grate.		Depth of Fire box. Front end.		Depth of Fire box. Back end.		Total No. of Tubes.	Length of Tubes.		Outside diameter of Tubes.
						FT.	IN.	FT.	IN.	FT.	IN.		FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.		FT.	IN.	IN.
14	96	Despatch Boat	Designed by Author	2	100	16	9	8	0	7	3	6	5	9	2	1	3	8	3	8	584	4	3	$2\frac{1}{2}$
15	103	H.M.SS. "Vigilant," "Wanderer," and "Osprey"	Messrs. Maudslay, Sons & Field	2	100	19	4	9	6	5	10	4	6	4	2	10	3	6	3	6	384	192 6 192 6	 6 0	$2\frac{1}{2}$
16	109	Gunboat	Designed by Author	2	100	19	9	12	0	5	9	4	6	0	3	0	3	3	3	3	780	400 5 380 7	 9 6	$2\frac{1}{2}$
17	115	H.M.SS. "Mutine" & "Cameleon"	Messrs. Maudslay, Sons & Field	1	66	9	8	18	4	7	6	4	5	10	2	0	3	10	3	10	262	6	0	$2\frac{1}{2}$
18	118	H.M.SS. "Ajax" and "Edinburgh"	Do.	2	450	34	6	15	6	9	10	12	6	4	3	0	3	6	4	2	1506	4	10	$2\frac{1}{2}$
19	128	H.M.S. "Niger"	Do.	4	400	16	0	11	0	8	9	12	6	4	3	0	3	7	3	10	1570	5	0	$2\frac{1}{2}$
20	137	Steam Yacht	Designed by Author	1	25	11	6	7	0	7	0	1	6	0	2	9	2	9	2	9	72	8	0	$2\frac{1}{2}$
21	140	H.M.SS. "Reindeer" & "Perseus"	Messrs. J. & G. Rennie	2	130	16	0	10	6	7	6	6	5	6	2	10	4	6	4	6	528	6	0	$2\frac{1}{2}$
22	141	S.SS. "General Murillos" & "General Victoria"	Do.	4	200	12	0	10	0	7	0	8	5	9	3	0	3	0	2	0	628	6	0	$2\frac{1}{2}$
23	144	Bombay Steam Navigation Co.	Messrs. James Watt and Co.	1	30	7	1	11	0	8	6	2	4	9	2	2	3	0	3	0	144	4	11	$2\frac{1}{2}$
24	146	S. S. "Tyrsaad"	Do.	2	120	8	0	15	0	10	0	8	7	0	2	0	3	0	3	3	520	5	8	$2\frac{1}{2}$
25	149	H.M.S. "Hornet"	Do.		100	16	6	9	6	7	6	6	5	3	2	4	3	0	3	6	440	5	6	$2\frac{1}{2}$

CHAPTER IV.

SUPPLEMENTARY DETAILS OF MARINE BOILERS.

THE CAUSE AND EFFECT OF SUPERHEATING.

To fairly understand any scientific question —whether relating to chemical compound or mechanical arrangement—a knowledge of the origin of the subject investigated is essential. In the present instance an addition to the marine boiler is introduced: and in common with other descriptions, a brief notice of its origin will not be out of place. The natural state of steam is undoubtedly a pure gas, but the mechanical contrivance for its production being at fault, a compound is formed, composed of steam and water. Now the presence of the latter greatly lowers the effect of the former, and it is from this cause that the steam is sometimes dried before it enters the cylinder. With marine boilers this is especially requisite, due to the steam spaces often being imperatively reduced in height above the water, from which the vapour is generated.

The authorities on superheating do not agree on all points as to its value in a commercial sense. Some advocate it as the best means to ensure the full effect of the steam. Others condemn it on account of the high temperature attained, injuring or defacing the surfaces in working contact. Now if the latter evil can be overcome, a great gain is accomplished, and repair thereby reduced. That, by drying the steam, it is rendered more powerful than when saturated, is an acknowledged fact, and perhaps no authority has proved this better than the eminent engineer, Mr. John Penn, who stated in 1859, when addressing the members of the "Institution of Mechanical Engineers," On the Application of Superheated Steam in Marine Engines:—

"The real source of advantage in employing superheated steam appears to be in preventing the presence of any water in the cylinder of the engine, and ensuring that the cylinder shall never be occupied by anything but pure steam; making it a real steam engine instead of one working with a mixture of water and steam. In all condensing engines the interior of the cylinder being open to the condenser during half the time of each revolution of the crank is in communication during that time with the low temperature of the condenser, or about 110° when the vacuum is 13½ lbs. per inch below the atmosphere or 27 inches of mercury. There is consequently a rapid radiation of heat from the sides and end of the cylinder, cooling down the whole mass of metal. The steam admitted into the cylinder in the next stroke, at a temperature of 260° if at 20 lbs. per inch above the atmosphere, coming in contact with these cooled

surfaces, heats them up again, being robbed thereby of a portion of its heat; and the consequence is the deposit of a quantity of water in the cylinder, from condensation of an amount of steam proportionate to the quantity of heat imparted to the metal of the cylinder. A portion of this water in the cylinder may be evaporated again into steam towards the end of the stroke, by carrying the expansion of the steam down to a sufficiently low pressure; but even then its effective value as steam in propelling the piston will have been lost during all the previous portion of the stroke. The engine must in fact be looked upon as only in degree better than Newcomen's atmospheric engine, in which the whole of the steam was condensed in the cylinder at each stroke; and the advantages of Watt's great invention of condensation in a separate vessel are not fully realised until this serious defect is removed. Now if as much heat be added to the steam by superheating it before entering the cylinder as will supply the amount of which it is robbed by the cylinder, it will remain perfect dry steam throughout the stroke, and not a drop of water will be deposited. This Mr. Penn believed to be the mode in which the superheating of steam acts in producing a saving of steam, and consequent economy of fuel, by preventing the extensive waste of steam that ordinarily takes place, and this indicates the extent to which the superheating can be carried with any great advantage. Mr. Penn also believes that an addition of 100° of heat to the temperature of the steam ensures the accomplishment of the desired object with steam at 20 lbs. per inch above the atmosphere, as used in marine engines. The steam is thus heated from 260° to a temperature of 360°, and is then only about as hot as the ordinary high pressure steam of 120 lbs. per inch used in locomotive engines.

"The plan of superheating the steam before entering cylinder is a simple and eligible mode of attaining the desired object, and appears also to be preferable to a steam jacket. For when the steam is supplied to the jacket from the same boiler as the cylinder, the supply of heat to the metal will be slower than in using superheated steam, owing to the difference of temperature being less; and to carry out the object fully requires the steam in the jacket to be superheated, and the cylinder covers to be also jacketed, since in the short-stroke marine engines where the diameter is nearly double the length of stroke the area of the two covers or ends equals that of the sides. But even then the application of the heat by the steam jacket is outside the cylinder, and the heat is delayed in its action by having to pass through the thick metal; whereas by the introduction of superheated steam into the interior of the cylinder the object is accomplished in the most direct manner, by heating the surface with which the steam comes in contact, and even a momentary chill of the steam down to the condensing point is entirely prevented. By superheating the steam with the waste heat of the smoke box, not otherwise usefully available, all this effect is obtained without cost; but with the steam jacket the heat used has to be supplied from the boiler. An important practical advantage attending the use of superheated steam is obviously that all objectionable joints for steam jackets are avoided; and the cylinder being felted and

lagged the same as the steam jacket, there will be no more loss of heat by radiation from the outside.

"The mode of superheating the steam may be varied in many ways, a general principle to be aimed at being to make use of the waste heat for this purpose after leaving the boiler, so as to accomplish the superheating without any cost of fuel, and to place the apparatus where it will not be exposed to injury from too great heat. The superheating apparatus has generally been placed in the smoke box or uptake flue in marine boilers, and has consisted of fagots of tubes or coils of pipe for the purpose of obtaining the required extent of heating surface within a limited space."

Reference was next made by Mr. J. F. Spencer, the kind contributor of plates Nos. 3 to 8, in this work, "on the variation on the temperature of steam in different parts of the boiler when superheated."—"He remembered noticing in the boilers of a large steamer, which had high steam domes with the uptake flue passing up through so as to superheat the steam at that part, that the temperature of the steam in the top of the dome was as much as 340°, whilst it was only 200° in the boiler just below the dome."

Mr. E. A. Cowper next observed, "That the pressure did not vary with the temperature; and whatever superheating took place, the effect could only be an increase in the volume of the steam and in its temperature, as it would be impossible for any difference of pressure to exist in the superheating apparatus, except indeed a slight diminution of pressure that would arise from the resistance of the small tubes to the passage of the steam. The first effect of the superheating would be the evaporation of all the moisture in the steam, as steam always left the water in a boiler in a more or less wet or damp state, from the mixture of minute particles of water with it, even when there was no sensible priming; it would then become perfect or dry steam, but at first would not be raised at all in temperature; but when the superheating was carried beyond that point, the temperature of the steam would be raised by all the heat added, and its volume proportionately increased, causing an increase in the total quantity of steam supplied at the same pressure and from the same evaporation of water. Steam was expanded by increase of temperature at pretty nearly the same rate as air and other gases; and since air at 32° was doubled in volume by an increase of temperature of 480°, steam at 20 lbs. per inch or 260° would be doubled in volume by 708° increase of temperature ($480° + 260° - 32° = 708°$); and a rise of 100° from 260° to 360° would consequently increase its volume 1-7th, causing an equal saving in consumption of fuel when the superheating was effected by using the waste heat of the smoke box. As the specific heat of steam was only about 3-4ths that of air, steam would require only 3-4ths the quantity of heat to be supplied to it to produce the same rise of temperature; and partly for this reason steam was now used instead of air in caloric engines, since the same effect of expansion was thereby obtained with so much less supply of heat.

"There was no doubt that in cylinders without steam jackets condensation of a portion of the steam took place at the beginning of the

stroke, and a partial re-evaporation at the end, on account of the metal of the cylinder being colder than the fresh high pressure steam entering from the boiler, but hotter than the expanded steam in the cylinder at the end of the stroke: since the whole metal of the cylinder could not change in temperature twice in each stroke (though the interior surface must do so), the temperature of the cylinder and piston must be an average of the temperature of the whole of the steam coming in contact with them. He had tried a direct experiment suggested to him by Mr. Appold, namely, fixing a glass gauge tube in communication with the interior of the cylinder, the outer end of the tube being closed. At the beginning of the stroke the interior of the glass became quite dull with moisture from condensation going on in the cylinder; but towards the end of the stroke the moisture was entirely evaporated and the glass became clear, showing that there was perfectly dry steam in the cylinder by that time. The cylinder was in fact a partial condenser at the beginning of the stroke, and a boiler at the end of the stroke; and if it were not for this boiling off of the condensed water at the end of the stroke, the cylinder would soon get very nearly to the temperature of the steam.

"In an expansion engine without a steam jacket he had found by a comparison of the actual indicator figures with the theoretical figures which ought to have been obtained if no condensation had taken place in the cylinder, that the loss of power when cutting off the steam at $\frac{2}{7}$ stroke amounted to a loss of 11·7 per cent; at $\frac{1}{5}$ stroke to a loss of 19·6 per cent.; at $\frac{1}{8}$ stroke to a loss of 22·2 per cent.; at $\frac{1}{11}$ stroke to a loss of 44·5 per cent.

"But when the cylinder had a steam jacket supplied with steam direct from the boiler, he found the actual indicator figure almost exactly corresponded with the theoretical figure, except that at the end of the stroke it was raised a little, about $\frac{1}{2}$ lb. in pressure above the theoretical line, in consequence of the superheating of the expanded steam from the higher temperature of the metal of the cylinder. With steam in the jacket of the same pressure as that in the boiler he did not think there could be any condensation in the cylinder; for all that was requisite to prevent this was to keep up the metal of the cylinder at the temperature of the entering steam, by supplying the heat abstracted by exposure to the cooler steam during expansion, and that lost by radiation, which was very small in a well lagged cylinder. The piston ought to have non-conducting surfaces or plates, and the cylinder ends should have steam jackets."

Mr. Penn also stated, when reference was made to the mixing the superheated and ordinary steam, that "the superheating apparatus must certainly be regarded as a portion of the heating surface, and some reduction might consequently be made in the boilers on that account, besides the reduction due to the saving effected in the quantity of water to be evaporated in doing the same work. In reference to the question of mixed steam, he had not tried any experiment on the subject, and did not see that as regarded the final effect it mattered how the steam was heated, provided the temperature of the whole steam that

entered the cylinder was raised the 100° required to prevent any portion of it being cooled below its natural temperature whilst passing through the cylinder; and it seemed to him that the best way of attaining that object was to heat the whole steam on its passage to the cylinder."

The very proper question was next mooted by Mr. Cowper, as to what lubricant should be used in the cylinders with superheated steam. He "considered that oil was not suitable for lubrication with highly superheated steam, as it was burnt up or charred rapidly at such high temperatures: 360° was certainly quite a safe temperature, and he thought even 400° might be made use of if oil were not employed for lubrication. He had known an instance where 100 lbs. steam was used in a double cylinder engine, expanded 12 times, and the high temperature of the 100 lbs. steam (about 340°) was kept up in both steam jackets; in that case there was found to be some cutting of the piston, but oil was used for lubrication. He had suggested a plan for preventing this by keeping the cylinders about 100° cooler than the steam, so as to cause a slight condensation upon the surface to keep it damp, and so prevent its cutting with the friction of the piston. He had tried an experiment which proved that even with very hot steam the surface could be kept damp if it were 100° lower than the temperature of the steam."

Soap and water have been proposed by some and, indeed, used as a lubricant, and also pure water, which in either case must partially condense the steam. With *pure* grease or oil as a lubricant no great effect is seen as to the cutting of the surfaces. One fact is certain, also, that with surface condensers, oil must not be used in the cylinder to any great extent. The author has had experience in this matter, and in one particular case, with superheated steam and surface condensers, no lubricant was used in the cylinders for several days and nights, during which the engines were in constant motion. On the ship arriving at its destination the surfaces were examined, and found to be nearly perfect as when new. Making further allusion to the temperature of the steam most practicable, the late Mr. John Elder, of the well-known firm on the Clyde, has stated 80° Fahrenheit above the natural temperature of the steam to be the most beneficial.

The lately lamented Mr. Langdon, formerly in the employ of Messrs. James Watt and Co., some time ago informed us that he had proved 290° Fahrenheit for the total heat of the steam to be the most agreeable temperature for the working surfaces in contact with the same, and one square foot of surface for the superheater per nominal horse-power he deemed sufficient.

It is obvious from the foregoing remarks that two distinct attainments must be accomplished to produce the required effect, viz., the requisite temperature and time for the steam to absorb the same. It requires no great research into the depths of science to understand that the time permitted for the absorption of caloric regulates the temperature of the steam, be it maximum or minimum. It is evident to all that the velocity of the passage of the steam in the superheater greatly determines the requisite amount of surface. Now from this simple fact being universal, various devices or rather arrangements for the distribution of the several gases

have been brought into practice. Apart from this cause for various productions, another fact must be noticed: the situation of the superheater is always in the smoke box or uptake. Here then is a practical recognition by the authorities of the imperfection of their boilers. Now were the combustion perfect, and the absorption of the caloric, by the water, more complete, the superheaters must be placed in or near the combustion chamber, rather than beyond it. As it is arranged at present the waste heat is utilised, and thus a gain, however, may be said to be effected. It must be strictly understood that no particular arrangement of details is alluded to, but rather that universally marine boilers are faulty, but from circumstantial causes rather than caprice of idea.

Superheaters, like all other human productions, partake of various shapes, and even contortions in some instances, to attain the requirements; cylindrical, vertical, and angular corrugated surfaces, tubes, spiral shapes and other fanciful arrangements have been proposed and introduced in practice. Notwithstanding all these mootings the few have had the preference, and the principal arrangements at present under favour are cylindrical and tubular. Each example has, of course, its merits and disadvantages, the latter bearing more reference to cost of construction than effectiveness, while access for cleansing is, in some cases, not so readily attainable as it should be. Of course those evils exist more or less with all mechanical contrivances, and the main attainment is generally paramount when the whole is under pre-notice.

The next subject requiring description is the arrangements of superheaters.

CYLINDRICAL SUPERHEATERS.

Simplicity of construction is akin to perfection of attainment, and it is, doubtless, with this in view, that the authorities in general are in favour of the arrangement illustrated by Fig. 26, on the next page. The plan and elevation represent the superheater fitted by Messrs. J. & W. Dudgeon in the mail steamer "Ruahine," the general arrangement of the engines of which are illustrated by plates 9 and 10. The cylindrical superheater now noticed is secured on the top of the uptake; the diameter of the smaller cylinder is four feet six inches, and the outer six feet, thus permitting a space of nine inches for the traverse of the steam. It is seen, in the sectional plan, that four divisions or partitions are inserted in the casing; one is solid throughout, two are open at the top, and one at the bottom.

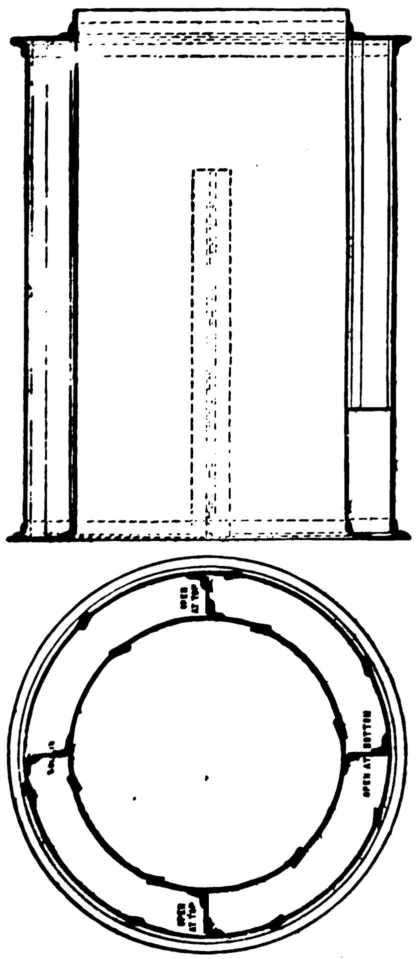
The action of the steam will thus be readily understood. On one side of the first or solid division is secured a pipe communicating with the steam space in the boiler—a stop-valve being there placed for use when requisite. The steam on entering the first space ascends to the opening at the top of the second divisional plate. The steam next descends and passes through the opening in the next divisional plate—opposite the solid plate. The line of progression is next to ascend and pass through the passage open at the top of the fourth divisional plate.

Situated on this same side of the solid plate is a discharge valve and pipe, the latter leading to the cylinders of the engines—the steam therefore descends a second time before leaving the superheater.

The action common with this arrangement is four traverses of contrary perpendicular or angular routes: first, ascent; second, descent; third, ascent; and fourth, descent. By this reverse travel of the steam time is allotted for the absorption of the heated products passing through the central portion of the superheater.

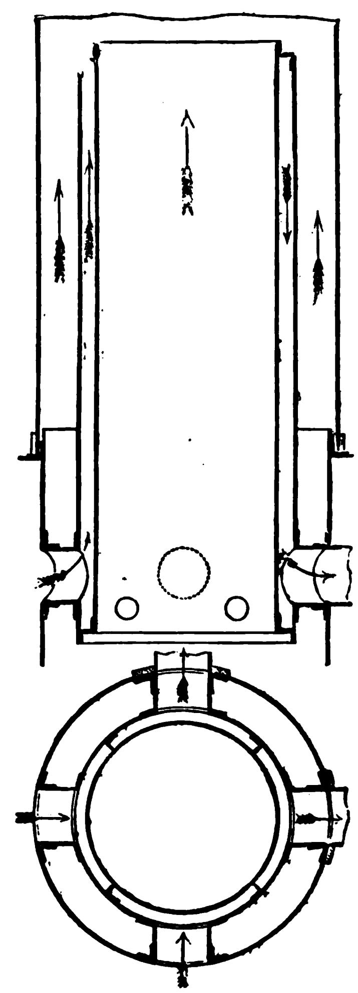
In other examples by the firm under notice, six divisional plates are introduced, four of which are open at the top, oppositely situated, and two open at the bottom, similarly arranged. The steam is therefore divided, and travels in opposite directions from the supply to the discharge. This arrangement is as that fitted in the S.S. "Mary Augusta," built and engined by the Messrs. Dudgeon. This firm are not isolated advocaters of the arrangement of superheater under notice. Other authorities patronize the scheme. Amongst the most prominent may be quoted Messrs. James Watt and Co., who were the originators of the arrangement under notice.

Fig. 26.



MESSRS. DUDGEON'S CYLINDRICAL SUPERHEATER.

Fig. 27.



ROWAN'S CYLINDRICAL SUPERHEATER.

It is seen that the volume of heated gases from the smoke box passes through the inside cylinder of the superheater, the outer casing receiving none of the effect. It is obvious that unless the casing is lagged or protected by some other means, a certain amount of caloric from the steam must evaporate in the surrounding atmosphere. This fact, doubtless, occurred to Mr. William Rowan, of the firm of Messrs. John Rowan and Sons, who, in the year 1859, patented the arrangement of superheater illustrated by Fig. 27, on the last page. As no person is presumed to understand any scheme better than the originator, a certain portion of the inventor's description is borrowed for the present purpose.

"This is an apparatus for superheating the steam after it has left the boiler, whereby the particles of water held in suspension in the steam are themselves converted into steam. For this purpose a cylindrical tube of metal is formed, and inside of this is placed another tube somewhat smaller, so as to leave an annular space between the two. These tubes are fixed in the flue or chimney of a boiler either for land or marine engines, and the steam from the boiler is caused to pass through the annular space, whereby it receives an additional charge of heat from the products of combustion conveyed through the inner and outside the outer tube in contact with its inner and outer surfaces.

"The steam issuing from the boiler through the divisional passages ascends to the top of the chamber, and passing over the partitions descends again to the bottom of the chamber, from whence it is conveyed to do its duty in the engine. The chamber being placed in the midst of the heated products in the chimney, both its inner and outer surfaces are heated and this heat is imparted to the current of steam circulating within.

"The apparatus thus far alluded to applies to cases where two distinct marine boilers are connected together, and having, consequently, two distinct steam pipes. In other cases, where there is but one steam pipe, two of the passages and the partitions are omitted."

Two of the partitions, opposite each other diagonally, extend the entire length of the annular chamber, or steam space. The others extend from the bottom for a given height, by which means spaces are formed from the top to the termination of the partitions for the traverse of the steam, or, to be concise, these partitions are open at the top, as those relating to Fig. 26.

The position of the pipes, from the boiler, is at right angles, and of course those for the discharge are similarly situated, the solid divisional plates or partitions forming the line of division.

The action of the heated products from the smoke box is on both sides of the steam casing, and thus surface is produced in preference to time for the drying of the steam; the latter being exposed to a double surface, or an internal and external current of heated gases, the preservation of the temperature is therefore maintained as much as the degree of heat is accomplished.

The line of the traverse of the steam within the annular space is thus. The four openings shown in plan common to each space between the partitions are, two for the supply, and the remainder for the discharge of the steam; the openings having the pipes connected are

for the latter purpose, or leading to the engines. On the steam being admitted into the allotted spaces, it ascends and traverses through the passages above the shorter partitions, and descends to escape through the pipes alluded to.

By this it is apparent a zigzag route is traversed by the steam of a single pitch, whereas with the previous superheater alluded to a threefold pitch is formed. In comparing these two forms of cylindrical superheaters, one point must not be lost sight of. In the first case, time for the traverse of the steam is the main agent to produce the same; while, with the present example, surface in contact with the heated gases is preferred to accomplish the same.

It next becomes requisite, when deducing the value of each arrangement, to consider the arrangements in a commercial sense. Granting that the steam can be heated alike with each example, the cost should be noticed. The arrangement shown by Fig. 26—page 158—is simply constructed, and the heated gases are not separated during ascent, consequently no relative proportion of divisional exit needs consideration. The means of supply and discharge are effected by two connections, one on each side of the solid partition. Now, with the arrangement illustrated by Fig. 27—page 158—more material and detailed construction are requisite than with that depicted by Fig. 26. It is noticed, also, that a duplicate set of pipes for the supply and discharge of the steam is introduced, to say nothing of the internal connections of the same.

Independently of the double surface exposed to the heated gases, the first example undoubtedly is the most economical, commercially, and, doubtless, by lagging the outer side of the casing, will be rendered equally effective as the second here brought into notice.

Next as to the access for cleansing, attention should always be devoted to this in all examples of mechanical construction. Many good ideas have met with condemnation simply from the neglect of the matter now under special allusion, and have been frowned at from this particular cause, by those practically acquainted with the subject.

The two examples now under notice being alike in principle of arrangement, the access for cleansing bear the same relation to each other; if there is any preference, the first example alluded to, perhaps, has the advantage of it.

TUBULAR SUPERHEATERS.

To be certain as to the value of any mechanical contrivance, the origin of the requirements should be duly considered. Now, with superheaters, two matters demand consideration; the first relates to the final result, and the second, the space and situation allotted for attaining the same.

What is required is a given amount of heating surface of metal in contact with the steam, in proportion to its volume and time allotted for its traverse. Now, this being the origin of the matter under issue, many tubular arrangements have been proposed, and, indeed, put in practice, all of course professing the most perfect result. Another cause for tubular superheaters is, that for war purposes all the fittings of the marine boiler —funnel excepted—must be below the line of

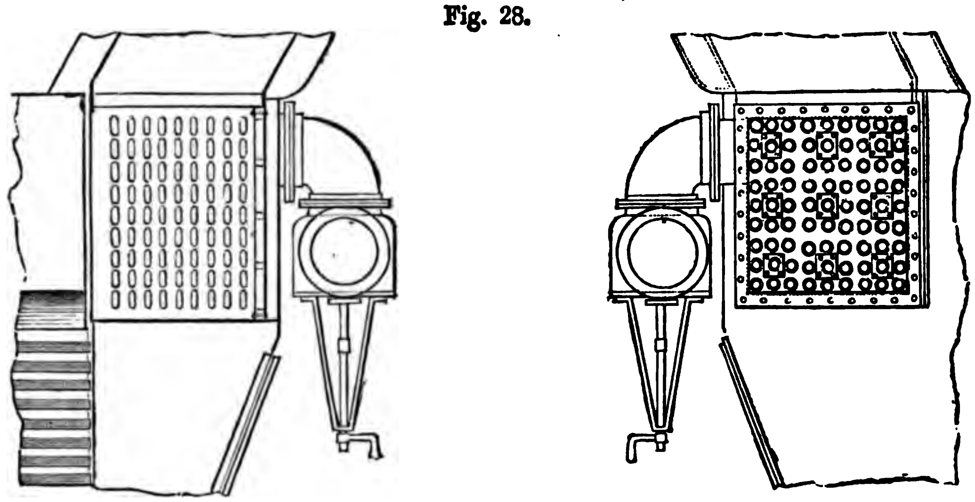
ion. Now, with regard to ordinary drical superheaters, this is next to an ssibility, hence the multitubular kinds been introduced.

ith all mechanical contrivances, construc- ınd its difficulties have to be considered e settling the question of utility Next, t prior to this, must be noticed the means cess for cleansing and repair, and, indeed, rangement of details is worthy of appre- n unless this matter is duly attended because nothing so effectually lowers the ion of the proposer of a novelty, than the ence of this omission in the scheme, which is not only indicative of an error of ght, but also an illustration of the absence actical knowledge.

exemplify: imagine a superheater with situated at angles with each other, and eat amount of heating surface is thus nable in the smallest compass. So far gain is effected, and the design almost ct. The flame, gases, and smoke operate he exterior of the tubes, and the steam s through them. Or, assume an undulated arrangement in the uptake of one continuous tube or piping so arranged that the products from the smoke box can surround it; the spaces between the tubes in each example to be about three inches, so that the arrangement is, therefore, compact. Now, while the surfaces in contact with the caloric are free from incrustations internally and condensed smoke on the exterior, all will be well; but when cleansing is requisite, the fallacy of the arrangement will be apparent.

With reference to repair: before knowing the requisition of the same, inspection is requisite, and, with a crowded cluster of tubes, this will be a matter of disarrangement, or, perhaps, worse still, the occurrence of a fracture will illustrate the desideratum, without permitting the prior inspection. From these remarks it is obvious that the principles already alluded to, when describing marine boilers and referring to the cause for given arrangements, apply in the present case, moreover especially relating to construction, inspection, and repair, the former being regulated by the two latter.

Fig. 28.



MESSRS. MAUDSLAY'S HORIZONTAL TUBULAR SUPERHEATER.

HORIZONTAL TUBES.

Amongst our late engineers the deeds of Joshua Field, Esq., C.E., are in the front rank of chronological records. That gentleman began early to put in practice his ideas on the subject of superheating, and indeed so successfully, that the firm of which he was a partner, Messrs. Maudslay, Sons, and Field, still continue to use the arrangement represented by Fig. 28—page 161. Mr. Field patented this in 1859, and describes the same in his specification nearly as follows :—

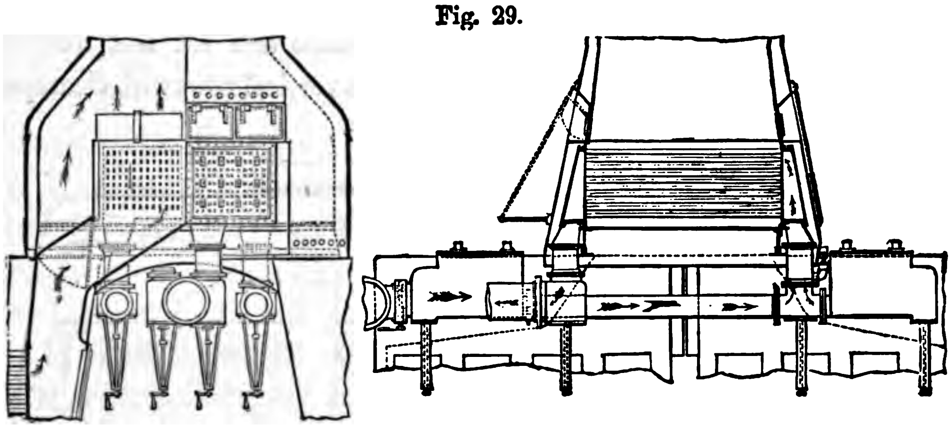
"This invention consists of an improved method of arranging and constructing apparatus for superheating steam, whereby the steam is made to pass through a system of flattened tubes, which are placed in the uptake in such a position as not to interfere with getting at the ends of the boiler tubes for cleaning or renewing them, and so as to offer the least resistance to the heated gases, and give the requisite area for their passage to the chimney, while at the same time the surface of the tubes is brought well into contact with the heated gases which pass up between them. The tubes are placed in rows one above the other, and have their ends fixed into steam boxes, one of which receives the steam direct from the boilers, while the other has communication with the main steam pipe leading to the engine. The ends of the flattened tubes are left round, and are fixed in the tube plates of the steam boxes with ferrules in the ordinary way, but as the flattened portion of the tubes will not pass through the round holes in the tube plates, the tubes are divided into groups of nine, as near as may be, and the holes in one of the tube plates for the central tubes of the groups are of such a form and size as will readily allow the flattened tubes to pass through them, and then bolt on to the plate small flange plates with round holes, into which the ends of the tubes are fixed in the ordinary way. By this arrangement any one of the surrounding tubes may be taken out and renewed, when required, by first removing the central tube of its group with the flange plate on its end, and then withdrawing the defective tube through the central hole made for that purpose. The sides of the tubes, placed in rows above one another, present a nearly flat surface, which may be readily cleaned with brushes through doors made in the chimney uptake; but in case where it is not convenient to clean them in that way, it is proposed to use a scraper worked from the outside for the purpose. Also, in cases where it is convenient to do so, it is preferred to provide additional passages for the heated gases, with the requisite dampers for shutting off the passages through the superheating apparatus, which may be used in case of the apparatus requiring repair, so that it may not be injured when it is found necessary to shut the steam off from it."

It is noticed that the top rows of the tubes of the superheater are below the roof of the boiler, an arrangement required in war steamers. The stop valves and pipes are arranged with the facility of either supplying the engines with steam from the superheater or from the boiler direct.

In cases where the superheater can be situated above the boiler, or for mercantile purposes, one apparatus is made common to the entire set of boilers. This will be readily understood

by referring to Fig. 29, which illustrates side and end elevations of the arrangement alluded to. One-half of the end elevation is shown in section, and the other with the cover of the steam box removed, in order to show the ends of the tubes. The damper is shown open to the superheater, and closed to the compartment at the side of it. The doors directly above the tubes are for access when requisite, to cleanse the exterior surfaces of the tubes. The arrangement of the stop valves will be understood by referring to the side elevation, to which attention must now be given. This pass into the boiler, by which means a lesser weight or load will be available. Presuming a plan of the arraugement of the valves and pipes, the latter, connecting the safety and stop valve casings, pass in a direct line in front of each boiler, and the supply branches connected to the superhcater are at the angles of the right-hand end. The discharge valves from the superheater are at the left hand, centrally of the space between the boilers, as seen in the end elevation. At the end of the left-hand valve box, a throttle valve is secured —seen in the side elevation. By opening the

Fig. 29.



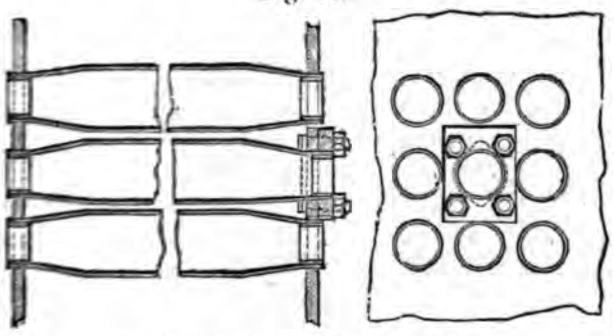
MESSRS. MAUDSLAY'S ARRANGEMENT OF STOP VALVES AND PIPING FOR SUPERHEATERS.

view shows the superheater and air casing in section. The safety and stop valve casings—one at each end below the superheater—are secured to the front of each boiler. A supplementary safety valve is situated between the right-hand main safety valve casing and the branch connected to the superheater. The use of this valve is to prevent the danger of explosion, should any moisture remain in the apparatus when the stop valves are closed. The discharge from this valve is—as shown—into the waste steam chamber of the main safety valve box; but it can be arranged to valve in question, steam can pass direct from the boiler, and by closing it the superheater can be brought into use. The arrows shown in this view indicate the passage of the steam when the throttle valves are closed. The products from the smoke boxes pass up through the superheater, as indicated by the arrows in the end elevation.

The means of cleansing can be better understood by alluding to the side elevation. Gratings are hinged for the men to stand on, one grating being shown in position for that purpose, and the doors above, at the ends of the tubes, are removed during the process.

With reference to repair, the access for this natural cause is one of the main considerations, and the lamented inventor was doubtless well

Fig. 30.



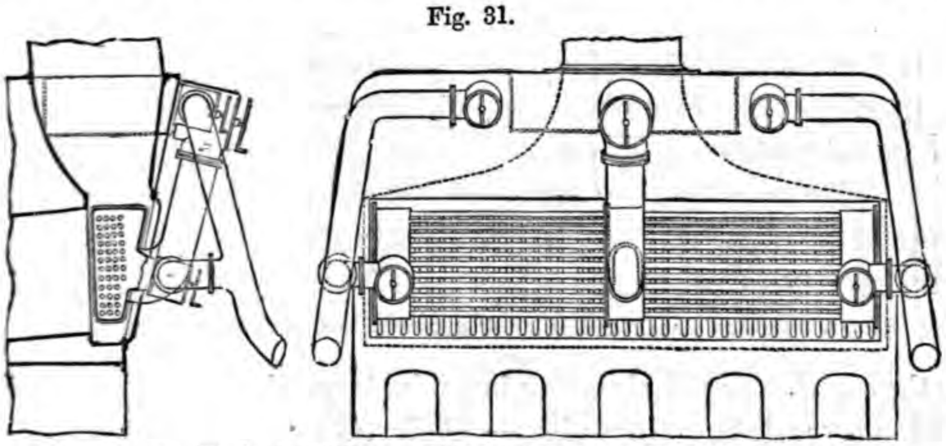
DETAILS OF MAUDSLAY'S SUPERHEATER.

informed in this matter, as the illustration, Fig. 30, portrays. This is an enlarged section and end view of nine tubes. It is noticed that the central portion of the tubes seems to be enlarged, but they are actually flat at this part, to cause the least resistance to the heated products from smoke box. Now it will be readily understood that, after the disconnection of the plate secured by the four bolts and studs, the tube direct can be removed, and those surrounding pass through the same opening if requisite. This, then, is one of the various attainments which the examples in question acquire.

The arrangement next to be noticed is that fitted by Mr. John Penn in the steam ship "Valetta," and described by him in his paper "On the Application of Superheated Steam in Marine Engines," as follows:—

"Fig. 31 shows the arrangement used by Mr. Penn, and employed in an extensive trial of the plan in the Valetta steamer of the Peninsular and Oriental Company, of 260 nominal horse power, running between Malta and Alexandria. In the smoke box of each boiler is placed two horizontal faggots of tubes, forming the superheating apparatus, each consisting of forty-four wrought iron tubes two inches diameter inside and six feet three inches long, placed in vertical rows with clear spaces between

Fig. 31.



MR. PENN'S HORIZONTAL TUBULAR SUPERHEATER AND FITTINGS.

them horizontally, for allowing ready access in cleaning the boiler; these spaces are left opposite each row of tubes in a tubular boiler, but in the present case the boiler is constructed with Lamb's vertical flues in place of tubes. The superheating tubes are fixed in the three flat chambers, which are made of wrought iron welded up at the corners and closed each with a single flange joint. The steam is supplied from the boiler to the centre chamber through

the stop valve and pipe, and is taken off from the end chambers by the stop valves communicating with the steam pipes leading to the engines. The steam is thus made to pass through the superheating pipes on its way to the cylinders, and becomes superheated by taking up a portion of the waste heat escaping from the boiler flues before reaching the uptake flue leading to the chimney. The steam pipes have also the ordinary direct communication with the boiler through the second stop valves, so that the whole superheating apparatus or either half of it can readily be shut off and disconnected at any time if desired.

"The vessel has made two trips from Malta to Alexandria and back, a total distance of 3276 miles, with the superheating apparatus; and then two of the same trips without the apparatus, but with no other alteration. The result was a saving of 20 per cent. in the consumption of fuel, although the men were not experienced in the management of the apparatus; and there appears every reason to believe that when the apparatus has been a little longer time in use the saving will be still greater. The main object kept in view in the detail of construction of the apparatus was to ensure a simple and durable plan that would not require any repairs for a long time; and for this purpose the superheating tubes were made a thorough mechanical fit, and free from strain of expansion tending to make them leaky. The wrought iron tubes are $\frac{3}{16}$ inch thick, and have thick ends welded on them; these are turned down to a square shoulder, and all correctly to the same gauge for length, and fitted tight into the holes of the tube plate, which is also planed on the face and accurately bored; the tubes are then pressed into their places all at once by the plates being drawn together with screws, and are made steam-tight by the fit alone; the ends of the tubes are then expanded. The total area of superheating surface including the wrought iron boxes is 374 square feet in each of the two boilers, giving a proportion of $2\frac{3}{4}$ square feet of superheating surface per nominal horse power, the engines being of 260 nominal horse power, and the boilers having a heating surface of 19 square feet per nominal horse power. This proportion appeared from Mr. Penn's trials to be sufficient for superheating the steam to the extent that is desirable. The apparatus has not leaked or failed in any way during the time it has been at work, and appears likely to prove very durable.

"The heat employed for superheating the steam is taken entirely from the waste heat after leaving the boiler, which would otherwise have escaped by the chimney; and this abstraction of heat from the smoke box, together with the screen of superheating tubes shielding the smoke box doors, has produced a marked effect in keeping the stoke hole uniformly much cooler when the superheating apparatus was applied than without it. The temperature of the steam is constantly indicated by a thermometer, which is fixed in a small cup projecting into the interior of the copper steam pipe, and containing a little mercury at the bottom in which the bulb of the thermometer is immersed. The fluctuations of this thermometer indicate very delicately the variations in temperature of the steam; and the mercury in the thermometer is affected considerably by the changes in firing, falling when the fire door is opened for fresh firing.

"In this arrangement no additional space is required for the superheating apparatus, the whole being contained within the ordinary smoke box, without any alteration of the boiler or any interference with its construction; the only external addition being the stop valves communicating with the apparatus. This apparatus can therefore be readily applied to ordinary marine boilers, without requiring any alteration beyond the extra connection and stop valves, and without interfering with any of the arrangements of the engines or boilers; and the important saving of 20 to 30 per cent. of the fuel can thus be effected, without incurring any risk of trouble or delay from the superheating apparatus. In case of any failure of the apparatus, it will be seen that it is only necessary to shut one set of stop valves and open the other."

Mr. Penn then observed "that the trial of superheated steam had been determined upon in the case of the vessel described in the paper, after the completion of the boiler; and the time being very short for the fitting up of the apparatus, he had to devise a means of accomplishing it without interfering with the work already done, and had consequently adopted the plan shown as the simplest arrangement and the quickest for construction. The apparatus was simply a work of repetition in the parts, the superheating tubes being all exactly alike, and fitted by machine work; the great object in view was to ensure against any risk of interfering with the efficiency of the vessel by failure or accident with the new apparatus, and to arrange the whole so that it could readily be disconnected, and the work carried on exactly the same as before the application of the superheating apparatus.

"He had not had an opportunity of trying any experiment with it himself, and did not consider the trial at present made a fully conclusive one as to results; but the vessel had been three months working since the apparatus was applied, part of the time without the apparatus for the purpose of comparison, and a pretty satisfactory proof of its success was that the engineers were very glad to get the apparatus in again; and there was found to be a reduction of 20 per cent. in the consumption of fuel when the apparatus was used. He had tried one approximate experiment with the apparatus before the vessel left this country, by graduating the opening of the injection cock of the condenser, and observing the extent of opening required for working with and without superheated steam; and he found that little more than two-thirds of the quantity of injection water was required when the steam was superheated, showing that a much smaller quantity of steam must have passed through the cylinders into the condenser, with a corresponding saving in consumption of fuel in the boilers."

This arrangement was not deemed perfect, it will be noticed, by the constructor, but rather as a combination of details to meet a certain purpose, and that the locality determined the design rather than the result of idea.

Analagous with the principle of the arrangement last alluded to is that represented by Fig. 32 —page 167. The sectional elevation shows the superheater longitudinally, the supply and discharge steam chests being in section. Instead of the superheater being directly in front of the tubes, as in Fig. 31—page 164,—it is preferred, in this instance, to fix it higher in the uptake, and thus receive the effect of the vertical action

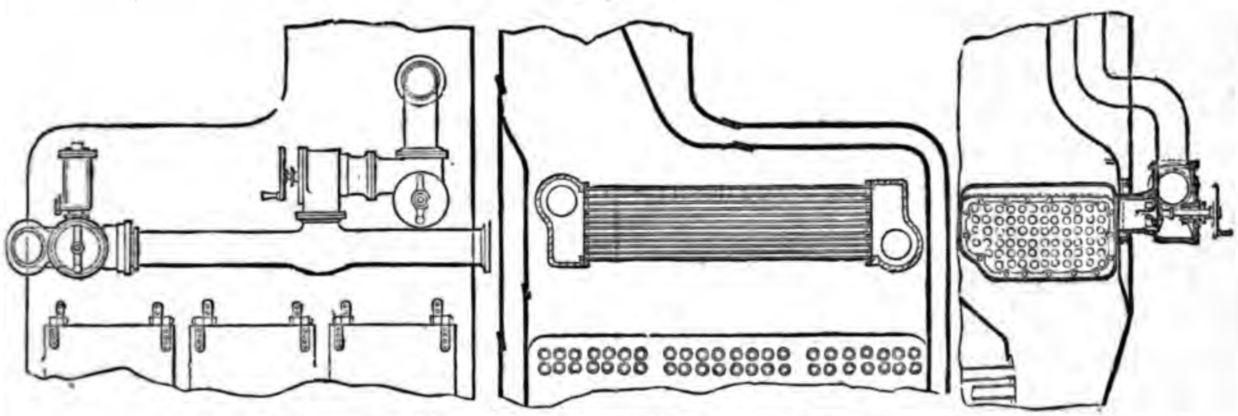
of the heated products from the smoke box. This is further represented by the transverse section; also the stop valve and supply pipe. It is obvious from these illustrations that the apparatus under notice consists of two chests, alike in form and dimensions, secured at each end of the smoke box. Those chests are connected by a given number of straight tubes, through which the steam passes direct from the source of supply to the discharge. The flame, &c., from the boiler tubes encompasses or passes amongst those of the superheater, and thereby dries the steam during its traverse.

steam from the adjoining boiler. The safety valve on the top of this casing prevents explosion in the superheater, from an excess of temperature and leakage.

In the event of the steam being required to pass direct from the boiler, or the superheater disarranged, the stop valves connected to the steam chests are closed, and that horizontally situated above the main pipe is opened.

This arrangement of the details is convenient for manipulation, and simple in arrangement. The exterior portions of the apparatus can be readily cleansed without its removal, but the

Fig. 32.



MESSRS. RAVENHILL'S HORIZONTAL TUBULAR SUPERHEATER AND FITTINGS.

In the front elevation a set of pipes and stop valves are shown, for the purpose of permitting or preventing the flow of the steam through the superheater. The arrangement can thus be described: the supply pipe is the highest connection on the boiler front—also seen in the transverse section. This pipe is connected to the supply stop valve casing in communication with the supply chest in the uptake. The steam, after passing to the discharge chest, enters the end stop valve casing, and thence to the boiler. The pipe, in connection with the stop valve casing last alluded to, admits the traverse of the

requirements for internal access are not equally convenient.

The well-known paddle despatch vessels, H.M.SS. "Helicon" and "Salamis," have fitted in their boilers an arrangement of superheaters similar to that now under direct notice, constructed by Messrs. Ravenhill and Hodgson. Other firms doubtless have adopted the same arrangement of details, with, perhaps, slight deviations, but not affecting the principle of the requirements.

It is obvious that the principle adopted to accomplish the same result is alike in each of

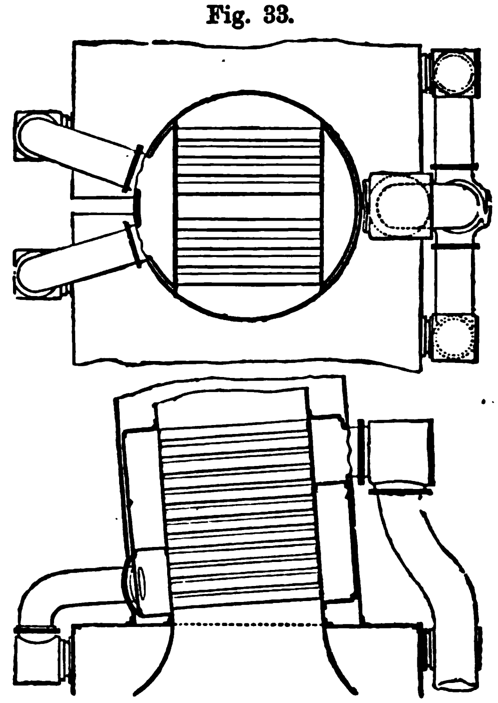
the four arrangements last alluded to. The saturated steam, by passing *direct* through the apparatus, depended on the surfaces in contact rather than the time occupied during traverse. Superheating depends on two causes, as before stated, to produce one effect. The engineer has two functions at his will—surface and time; and his apparatus must be arranged to employ either or both simultaneously. It may be argued, that with time as the main agent, friction will be involved, and a greater bulk of steam therefore requisite in the boiler to act as propulsion than when surface alone is depended on. Now, admitting that retardation of the volume during traverse can only be produced by intricacy of passage, we will give a brief notice to the friction involved with the arrangements last under notice, before describing further examples.

The steam, on entering the supply chests, forms a bulk equivalent to the contents of these compartments. Now the tubes are the means of passage, and the area of each, minute, in comparison with that of the supply steam pipes. It is certain that the steam, on entering the tubes, must conform to their several areas, and thus a given amount of friction the result. Here is a practical exemplification of the fact, that although simplicity of passage is preserved for the steam, friction is produced before the volume commences to traverse through the apparatus; retardation, therefore, is effected at the place least requiring it. Of course, a certain amount of the proceed from the smoke box is in contact with the chests in question, but the principal bulk passes amongst the horizontal tubes.

The matter next to be considered is whether, setting aside construction, it is not more analagous with natural laws to depend on time to superheat the steam than surface. Of course, these functions are quoted figuratively at present.

It is current amongst all interested in these matters that, if the steam is *retarded* during its traverse through the passage whose exterior is surrounded by the heated gases, that attainment is equivalent to a greater amount of surface, combined with a quicker means of traverse or a direct flow. It is with the knowledge of these

Fig. 33.



INTERNAL TUBULAR SUPERHEATER.

facts that the above arrangement, represented by Fig. 33, was produced. This apparatus is a cylindrical vessel secured on the roof of the boiler, having within it three compartments, the central containing the superheater. The sectional plan shows the divisional plates,

forming the supply and discharge chests, and the passage for the heated products from the smoke box. The supply steam valve boxes and pipes are on the left hand side, each connected to the respective boiler, two only being represented. The discharge valve boxes and pipes, on the opposite side, are situated to supply the engines with steam direct from the boilers, or through the superheater, as may be required.

As the sectional elevation imparts a more explicit view relative to the traverse of the steam than the plan, to it attention must now be given. The superheater, it is seen, consists of a series of tubes horizontally—or nearly so—placed in the compartment, through which the heated gases escape; and the supply pipes are at the base directly above the roof of the boiler, and the pipe for the discharge near the top of the apparatus. Two divisional plates or partitions are secured directly over and under the supply and discharge passages, thus forming four compartments vertically. These compartments, it will be noticed, are adversely arranged; those less in depth being at the top and bottom of the apparatus. The traverse of the steam consists of three flows between the supply and discharge passages. On the volume entering the lowest compartment it passes direct to that opposite, and the increased height of the same permits a return action through the tubes into the third space of similar height, from which a third traverse ensues into the fourth or last space.

By this arrangement longitudinal traverse is condensed into one-third of that required for a direct flow. The volume also is subdivided during its passage through the tubes, so that surface for the contact is attained, and combined with time to accomplish what is required. This arrangement, although shown on the roof of the boiler, can be secured below the same if requisite.

The means for cleansing has not been overlooked: it is noticed in the plan that spaces are left at the sides and centrally of the cluster of the tubes for that purpose, so that a brush or scraper can be readily manipulated when required. The internal surfaces of the tubes are accessible by the removal of doors—the latter not shown—at suitable localities on the casing containing the apparatus. This arrangement is designed by ourselves; but, while acknowledging it, we are cognizant of the facts that many firms have adopted the same in principle if not analogous in design.

VERTICAL TUBES.

The accumulation of sediment on the surfaces of superheaters is a subject demanding some attention from the marine engineer. Indeed, so far is this certain, that construction and design often give place to access for cleansing. The flame and smoke leave a refuse of easy removal, but not so the steam. This volume often defies the skill of the chemist as well as the engineer to remove the scale reposed during the passage of the vapour.

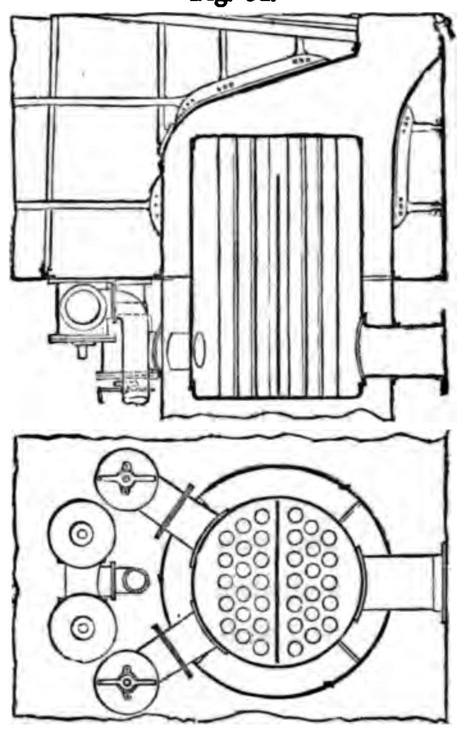
It will be noticed that the previous arrangements of tubular superheaters admit the steam through the tubes, and the volume therefore acts on the internal surfaces. Now the next consideration, akin to the positions of the details in question, is the means provided for removing the sediment from the steam accumulated within the tubes, and the incrustation from the smoke and heated gases on the external surfaces. The means for seeing can

assist the operation of cleansing the sediment from the exterior portions of the apparatus, but those internally, particularly the tubes, are cleansed by guesswork rather than with certainty. Notwithstanding these facts, the arrangements in question have met with much attention, while practical skill has been expended on them to attain the effects already produced.

Next as to the means for cleansing, attention should be given to the cause for repair. An arrangement may be a fair example as to design, construction, &c., but these attainments will, perhaps, accelerate repair rather than retard it. This is more apparent where the operation of cleansing is permitted to be of minor importance; the result being that the tubes are burnt through, and the superheater rendered a greater evil by its presence than a fault by its absence. Now it was these conclusions, probably, which practice exemplified that originated the vertical tubular arrangement. An example of this is shown by Fig. 34. On noticing the sectional elevation and plan, it is seen that the tubes are vertically secured within a cylindrical shell, and the latter is secured within the uptake, as low as the form of the same will admit. The heated products pass *through* the tubes, and also surround the shell, the uptake forming a casing for that purpose. The supply valve boxes are secured on the roof of the boiler, and the pipes are connected separately to the side near the top of the superheater. The discharge pipe is on the opposite side, of an area equivalent to those for the supply. The discharge stop valve, not shown, is secured at the front of the boiler.

It will be noticed in the plan that the tubes are separated centrally by a divisional plate; also, on alluding to the elevation, that the plate in question does not extend to the bottom of the apparatus.

Fig. 34.



CYLINDRICAL BOX VERTICAL TUBULAR SUPERHEATER.

The traverse of the steam, on entering the cylindrical shell, descends to the bottom, and after passing through the space allotted, the volume ascends to the discharge pipe. This arrangement produces a single undulation for the action of the steam, or, in common with prior examples, traverse of a single pitch.

It can be understood from the description here given relative to the present example, that the internal and external surfaces are exposed to the heated products and steam. It is obvious, also, that if there is any gain by the observation of time and surface for the traverse of the respective volumes, the arrangement in question partakes of the benefits to be derived. The

dispersion of the proceed from the smokebox has, of course a direct flow, while the steam is retarded to a certain extent.

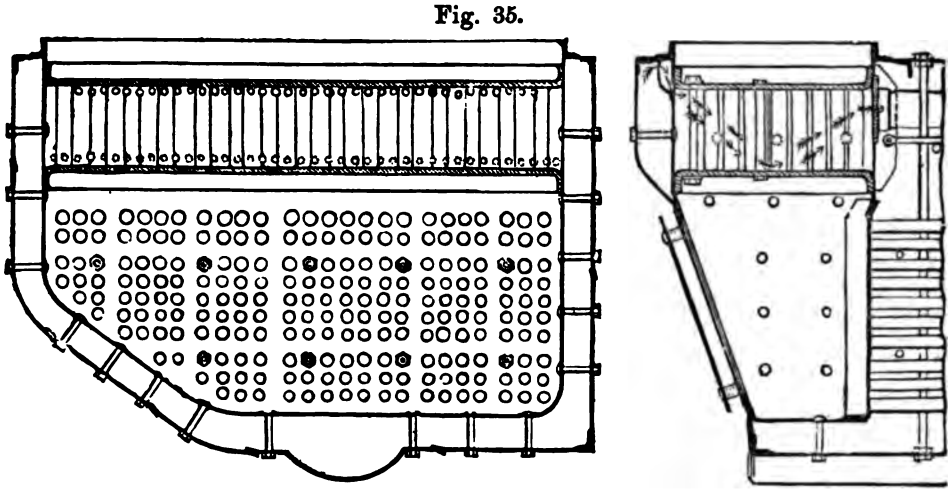
Next, with reference to the means for cleaning the apparatus in question. The surfaces in contact with the heated products are readily scraped or brushed, but those inside the shell are not so accessible. As there are no provisions shown in the illustration, a brief description is requisite. The annular space surrounding the shell is increased, at the sides, in a line with the divisional plate between the tubes, by the curved form of the uptake; and at these localities doors can be secured near the bottom of the shell. These doors being opposite each other, investigation is permitted, and thus a certainty as to the actual state of the surfaces made known. It often happens, however, with superheaters, that in the case of internal repair, their entire removal is requisite, and thus cleansing readily accessible. Notwithstanding this, means for inspection should always be provided.

The supply valves and boxes for the admission of the steam to the engines, independently of the superheater, are not represented, being beyond the limits of the illustration under notice.

This example has had a fair trial in the mercantile navy, and the result has been satisfactory, but not so far excelling others to warrant its universal adoption. It may be added, in conclusion, that the principle of this arrangement is analogous with that represented by Fig. 27, shown on page 158.

The examples brought into notice have to the present been shown fitted with valve boxes, and the several details requisite for the independent use of the steam.

Fig. 35.



MESSRS. HUMPHRYS' RECTANGULAR VERTICAL TUBULAR SUPERHEATER.

Amongst our eminent marine engineers, the late Mr. E. Humphrys claims notice for his mechanical productions, especially relating to the unbroken surface of the exterior; for example, his marine engines represented by Fig. 6, page 52. Analogous with this as to simplicity of arrangement is his marine boiler illustrated above by Fig. 35. This example represents an apparatus for superheating steam, patented by Mr. Humphrys in 1860. In the specification it is stated that " the invention has for its object

the improvements in steam boilers, when such boilers are employed to supply superheated steam, and consists in constructing the uptake of the boiler in such a form as to render it suitable for the reception of a number of vertical tubes, the uptake being part of the main boiler. These vertical tubes being introduced, serve as a superheating apparatus, and render it unneccessary to employ any pipes or valves for conducting the steam to such apparatus, and hence there is less liability to derangement."

The illustration now alluded to shows part of a boiler particularly applicable for ships of war, or of the types often termed as low boilers. The transverse section shows the evaporative tubes in the ordinary position, and the smoke-box beyond the same of the usual practice as to form. Directly above these tubes is secured a plate, and near the roof a second. These plates are connected by tubes and stays as represented. The traverse of the steam is indicated by arrows. It will be seen that a small compartment is formed near the roof, through which the first arrow is presumed to be entering; this means of entry is a series of holes. The passage of the steam into the superheating compartment is effected by similar perforations in the sides of the plate shown in the sectional elevation. The arrows in the transverse section, it will be noticed, are shown descending towards the centre, where a divisional plate is secured, perforated at the bottom, through which the single curved arrow is seen. These holes are shown dotted in the sectional elevation. The ascending arrows indicate the traverse of the steam to the discharge pipe, secured to the back of the superheater, directly below the roof of the boiler.

It is evident that the flow of the steam in this arrangement is retarded, and that time has been considered as one of the agents common with the cause required. The steam acts on the central portion of the bottom plate of the compartment, and therefore the top portion is rendered the *least* effective. Now there is a just reason for this: the volume from the smoke box acts on the exterior of the base plate of the superheater, and thus that surface is the *most* effective. The temperature of the heated products is the highest when entering the superheating tubes, and therefore the lower portions should be in constant contact with the steam. The principle of this is common with that relating to Fig. 34, shown on page 170.

The access for cleansing the surfaces in contact with the flame and smoke are amply provided, but the internal requisitions are not apparent. The provisions requisite for an independent traverse of the steam also are not shown, but doubtless the inventor has not overlooked this essentiality in practice.

When the arrangement in question is required for high boilers, the compartment near the roof is extended in depth to the bottom of the superheater, and the perforations are central of the height of the same, a direct flow only being maintained for the steam into the discharge pipe; the latter in this case being at the front of the boiler instead of inside.

Other firms have used vertical tubes for the purposes under notice, amongst which may be mentioned Messrs. J. Penn & Son, Messrs. James Watt & Co., and the Messrs. Rennie; the author also designed the arrangement illustrated on pages 83 and 93, by Figs. 12 and 13. On referring to this example it will be noticed

that the sectional elevation shows the superheating tubes vertically arranged in the smoke box; and on alluding to the front elevation, the extension of this compartment will be seen. It will be noticed also that at the top left-hand corner of the boiler front a stop valve box is secured, having a vertical branch; this latter portion is connected to the supply pipe, which passes over the superheater to the roof of the boiler. At the bottom corner of the sectional portion of the superheater, another stop valve box is secured—but not shown—to permit the discharge of the steam from the superheater. It is obvious from this description that the steam enters the superheater at the top at one end, and discharges at the bottom at the other, and thus a regular distribution of the steam is effected. Perhaps the better position for the supply valve is at the side of the boiler, or centrally of the end of the superheater. The position of the valves and pipes, as represented, renders certain access for manipulation within the boundary of the stoking room, in front of the boiler; a matter of the utmost importance with vessels intended for war purposes. The stop valve in connection with that supplying the superheater permits an independent flow of the steam to the engines, in which case the valves previously alluded to are closed.

The passage for the volume from the smoke box is of course enlarged at the uptake end, and directly above the termination of the tubular compartment a door or damper is hung, also one at the uptake. These dampers regulate the means of exit, whether through the superheater or the uptake. On closing the damper at the uptake and opening the one above, the heated products pass through the tubes of the superheater; and on reversing the position of the dampers, the apparatus in question is rendered non-effective.

The access for internal cleansing is not deemed requisite in this instance, but the accumulation from the smoke, &c., is readily removed both from the bottom plate and the internal surfaces of the tubes.

SHEET FLUES.

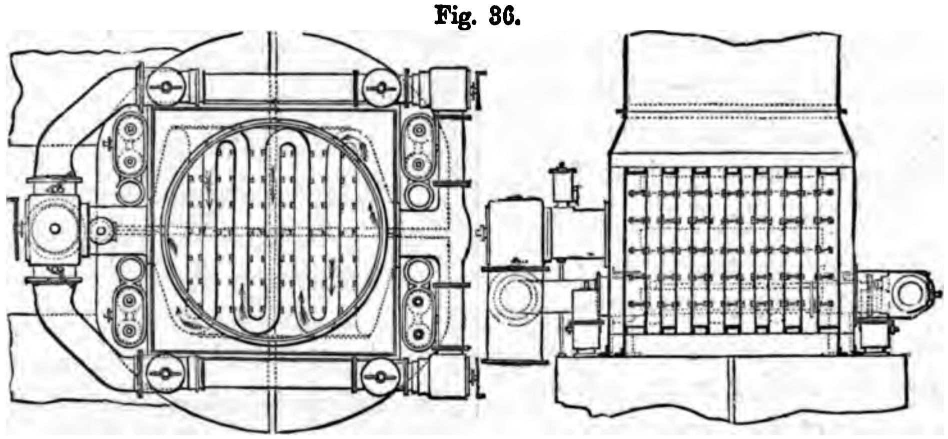
The adoption of surface and time combined with superheaters must be correct in principle and efficient in practice. A certainty of this is rendered more obvious when it is remembered that with the use of either function separately, the same result can be obtained. From the number of examples already noticed, it is apparent that superheaters have had their share of thought bestowed on them, and that each authority has contended with his brother in honest rivalry. It will be difficult, perhaps, to point out a more striking example, contrasting with those of prior notice, than that illustrated by Fig. 36 on page 174.

The sectional elevation presents a similarity to that of a series of elongated boxes vertically secured in the uptake. On referring, however, to the plan, it will be seen that the apparatus actually consists of an almost *zigzag* formed flue, through which the steam passes, as indicated by the arrows. Now to render this peculiar arrangement not only to be understood but to be appreciated, the following description is requisite. The largest circle in the plan denotes the funnel and the square form the casing encompassing the superheater, and those portions are also shown in the sectional elevation. The four stop valve boxes—one at each corner

of the casing—are connected to the roof of the boiler. The two boxes horizontally situated admit the steam to the superheater when requisite. The volume, after traversing through the undulated passage, reaches the main discharge valve, which latter, when opened, permits the flow to the engines. It will be noticed, on further examining the plan, that on each side of this valve box, last alluded to, a handle is shown, being a connection for manipulating the throttle valves, which latter are closed when the steam is passing through the apparatus in question. Now, assuming the steam is required to deemed compact, in common with the principle exemplified. The heated products from the smoke box pass direct through the intermediate passages, and the cleansing requisite, therefore, rendered a simple matter. The arrangement in question is a modification by ourselves of that recently fitted in some vessels belonging to the P. & O. Company's fleet.

The description of the several examples herein brought into notice is now concluded, and a brief summary of the several requirements common with the principle under notice will not be out of place.

Fig. 36.



INTERNAL VERTICAL SHEET FLUE SUPERHEATER.

flow from the boilers direct to the engines, the valves horizontally situated must be closed, also the main discharge valve at the left hand. The valves directly connected to the boiler are opened, also the throttle valves, and thus the superheater is closed to the steam.

It is seen both in plan and elevation that a supplementary valve box is secured on the main discharge branch pipe, being for the purpose of preventing disaster from an extreme pressure in the superheater. The main safety valves are similarly arranged on each side of the casing; and the entire arrangement can be

The engineer has to deal with several matters when he undertakes the cause and effect of superheating. First, he has steam in a saturated state; second, the heated products from the smoke box as an antidote for the bane; third, the application; and fourth, the form and construction of the apparatus.

Enough has been stated and illustrated to portray the several ideas generally entertained on the subject; and the matter in question is, therefore, now condensed direct to the principles most applicable for development. It has been clearly shown that two functions are avail-

able—surface for the contact of the heating volume, and time for the steam during passage. Of course it is obvious, also, that surface in either case is equivalent to time as far as the final result pertains, but the main question is the most economical means. Granting that the impact of the flame and heated gases on a given surface will produce a certain temperature, and that the steam will thus be dried within a short traverse, it becomes a matter of consideration as to whether repair will not be involved. Further than this, it is not erroneous to assume that separate surfaces, or, perhaps, both equal, should be provided for the contact of the steam and the heating products. It must be remembered that with superheaters an apparatus is formed, the surface of which is in contact with a powerful destructive element, and that the counteracting is of the feeblest kind. The matter therefore resolves itself into the fact that, to retain durability, a gradual flow for the steam should be maintained, and a direct traverse for the heated volume from the smoke box.

The engineer, as a rule, should not concentrate his thoughts to the accomplishment of a single purpose. Three essentialities should ever be before him, which are—effect, construction, and repair. A true recognition of these cannot fail to exemplify that theory and practice are naturally firm adherents in all matters of science, and in none more than that relating to marine engineering.

It has often been quoted that the individual who bears up manfully against any calamity is worthy of honour equally as the man who gains a victory. Now the engineer has much to contend with relative to marine boilers, and, to a certain extent, many of the difficulties have been overcome by his untiring perseverance. Sometimes it happens, however, that the suppression of one matter at fault causes the formation of another error, and thus patience and skill are sorely tried. The profession of an engineer who devotes his attention to marine matters is anything but a satisfactory pursuit to the practitioner. No matter how fortunate the result of his endeavours, much often remains to be overcome. What is at one time deemed a triumph, will at another be condemned, not, perhaps, by the absence of correct principles, but from the cause and effect of corrosion, either produced by sediment or galvanic action; and therefore the marine boiler can be truthfully termed the most troublesome portion of the causes to produce the effect of steam propulsion, and is a truthful illustration of the preceding remarks, as the sequel will clearly portray.

CAUSE AND EFFECT OF INCRUSTATION.

In dealing with the present portion of the subject now directly under notice, first, careful investigation must be made as to the cause for its presence; and, secondly, the most efficient means yet put in practice for its prevention. It is well known, of course, that the water with which the boiler is supplied is taken from the river or sea, as the locality of the vessel's destination determines. Now ocean traffic being by far the most important, it will be well to consider the properties of the fluid forming the sea. From a careful analysis made by Dr. Schwitzer (late of Brighton), of the compound parts of the sea in the British Channel, the following was the result (the relative value of the total quantities being 1000·):

Water	964·745
Chloride of sodium ...	27·059
Chloride of potassium ...	0·766
Chloride of magnesium ...	3·666
Bromide of magnesium ...	0·029
Sulphate of magnesia ...	2·296
Sulphate of lime ...	1·406
Carbonate of lime ...	0·033

Traces only of iodine and ammoniacal salt. The specific gravity was proved to be 1·0274 at 60°. It is obvious from this that the soluble matter contained in the fluid tested was $\frac{1}{28 \cdot 333}$ of the total bulk, and that chloride of sodium predominated in quantity.

Thus far an acquaintance is made with the soluble matter contained in sea-water when comparatively cool. Now as the fluid in question when brought into use undergoes a change from a liquid to a vapour or gas, it is obvious that the ingredients alluded to must be retained in some form during the transformation. Apart from this, attention must be given to the temperature, which, when the water is above boiling point, is common with the pressure of the steam. Next, consideration must be given to the nature of the solid matter accumulated; and, finally, the best means to remove it. That accumulation of substance is certain to be formed when the boiler is fed with water either from the river or ocean, needs no comment, practice having long ago set aside all doubts on that matter.

The original mode of disturbing the sediment on the internal surface of marine boilers was by repeatedly blowing out from the bottom of the shell. The gain effected by this was derived from the idea that, as the soluble properties of the water were naturally heavier than the fluid, an accumulation at the bottom of the boiler must be certain; and that, on causing a means of discharge, the internal pressure would effectually disturb the sediment. Practice proved, however, that a deposit remained on the upper surfaces, and thus the blowing out was not universally efficient. Next came the idea of surface blowing off; this, it was argued, must be correct, for as the ebullition occurred, the particles in the water would float, therefore a froth or brine be formed for a time; and considering the advantage of this, *scum* and *brine* troughs were introduced. After these trials as to the best means to be adopted, the natural conclusion arrived at was, that, to truly and effectually overcome the evil in question, the cause for its presence must be noticed.

In 1864, Mr. James R. Napier imparted much reliable information on the present subject, not, perhaps, as a conclusion for future progress, but rather as a result of actual experiments, in which facts were made apparent with doubt. In a paper delivered to the Institution of the Engineers of Scotland, in the year alluded to, Mr. Napier stated, in adverting to his previous belief in the gain by "blowing off:"—

"Believing, as I then did, in the ordinary theory of blowing off from the boiler before the water became saturated with salts, that an abundant feed and blow off would prevent the lime depositing, and therefore prevent the incrustations; and being desirous of saving the heat which would otherwise be lost by the great amount of blow off which I believed to be necessary, I had a regenerator made for the

S. S. "Lancefield" with about ten times the surface which it had been customary to give to such apparatus; but the results, as stated at a recent meeting, were so much at variance with my understanding of the ordinary theory, that I think a statement of the facts will help others to a clearer knowledge of the matter.

"The vessel sailed from Glasgow about noon every Thursday for the Hebrides, lay in one of the lochs there from Saturday evening till Monday morning, and arrived again in Glasgow on Wednesday, to recommence on Thursday a similar voyage. The steam was up or at hand all the voyage: about fifteen stops, of two or three hours each, were made each week, during which time the boiler was supplied with feed by a Giffard's injector, but little or no blow-off. While steaming, however, the quantity of water continually discharged through the regenerator was so great that the glass hydrometer used for ascertaining the density showed very little difference between the sea and the boiler water. The boiler was worked in this manner for about four weeks, and then examined; when, instead of being found, as I expected, clean, with little or no scale or deposit, the coating was much thicker than usual, but soft, very much like newly-made mortar, not difficult, before getting dried, to scrape off at all accessible places, but which when dry was nearly as difficult of removal as the ordinary compact scale.

"During one of the voyages, when I was personally directing the experiment, and had for some time been keeping the greatest amount of feed on the boiler which the engine could supply, I observed the water in the gauge glass got muddy, but did not then discover the cause. About two years before this I found the same soft limy deposit in the S. S. 'Isleman's' boiler, when trying similar experiments on the same station; but as I had disregarded the 'Isleman's' experience, and did not then know the experience of others, the 'Lancefield's' regenerator was continued, but with lesser quantities of feed and discharge, for about six months, when the tubes of the apparatus giving way, it was discontinued. I was fortunately saved further trouble, and the expense of repairing it, by discovering in the *Annales des Mines* for 1854, an interesting paper by M. Cousté, 'On the Incrustations of Boilers.' He there shows that the sulphate of lime can be deposited by heat alone, without any evaporation, and that at a temperature of 124 deg. Cent., or two atmospheres of pressure, sea water in its natural state is very near the point of saturation. As the 'Lancefield's' boiler was loaded to nearly 40 lbs. of absolute pressure, and worked generally at about two atmospheres, or 255 deg. Fahr., or about the point of saturation of the lime, it is clear that the greater the amount of sea water supplied to the boiler, the greater would be the quantity of lime deposited in it. And although there was a constant discharge from the surface by a conical tube, only some of the deposited matter —that which had not attached itself to the boiler—could be so discharged. If this be not the true explanation of the great deposit in the 'Lancefield's' boiler, of the difficulty of working boilers with sea water at higher pressures, and of the ordinary experience that boilers are cleaner when worked at a greater density, it will remain for others to explain it.

"M. Cousté's experiments, however, appear to me to be conclusive. He suggests a method

of getting quit of the lime by filtration at a high temperature. The following extract from his paper shows the conclusions he arrived at from his experiments:—

1. "'The sulphate of lime is less soluble in hot than in cold fresh or sea water.

"'For temperatures above 100 deg. Cent. the solubility of the sulphate of lime in sea water diminishes nearly in proportion to the increased temperature; and, consequently, this solubility diminishes very rapidly with the corresponding increase of pressure.

"'The following table indicates this solubility for different temperatures, as well as the degrees of concentration at which the saturation of sulphate of lime has place:—

Degrees of the Areometer corresponding to the Saturation.	Temperatures.	Pressures in Atmospheres.	Solubility or Proportion of Sulphate of Lime in 100 of Water at Saturation.
	Degrees.	Atmospheres.	
12½	103·00	1	0·500
12	103·80		0·477
11	105·15		0·432
10	108·60	1¼	0·395
9	111·00		0·355
8	113·20		0·310
7	115·80	1½	0·267
6	118·50		0·226
5	121·20		0·183
4	124·00	2	0·140
3	127·60		0·097
2	130·00	2½	0·060
1	133·30		0·023

"'This table expresses that, for example, sea water boiling at atmospheric pressure, or 103 deg., will arrive at saturation of sulphate of lime when it will have acquired the concentration of 125 deg. of Beaumé, and then it will contain 0·500 per cent. of this salt; at 1·25 atmospheres, or 108·6 deg. of temperature, the water will be saturated with sulphate of lime, when the areometer marks 10; it will then contain 0·395 per cent. of sulphate of lime; at two atmospheres, or 124 deg. of temperature, sea water in its natural state, and before it has experienced any concentration, is very near the point where the saturation takes place—for the natural water marks from 3 to 3·5 deg.,—and in this case the saturation takes place at 4 deg. of concentration.

2. "'Sulphate of lime becomes wholly insoluble either in sea water or in soft water at temperatures comprised between 140 and 150 deg. Centigrade; and if we expose at these temperatures water containing some of this salt in solution, it is entirely precipitated in the form of little crystals, or of very thin pellicles, according as the salt is more or less abundant in the solution. The sulphate thus precipitated is redissolved after the cooling, but as much more slowly as the temperature at which it is deposited is elevated. That which is deposited at 150 deg. takes many days to redissolve.'"

From this, the certain conclusion is that "surface blow off" is not deemed effectual, for Mr. Napier states from actual experiment, that "although there was a constant discharge from the surface by a conical tube, only some of the deposited matter—that which had not attached itself to the boiler—could be so discharged.

From Mr. Elder, in the discussion, some practical information was elicited.

"He stated he had a good deal to do with the working of boilers at from 30 lbs. to 35 lbs. pressure without surface condensers, and in some cases he had seen very extra-

ordinary deposits. One naturally expected to find most deposit in the section of a boiler where there was most salt and lime; but in a boiler divided into eighteen parts he had found, to his surprise, that although in the last section there were two and a half-times more salt in the water than in that of the first section, yet he could ascertain little difference in the quantity of deposit of lime in any section. He had therefore come to the conclusion that the deposit depended upon the temperature of the water, and not upon the quantity of lime in it. The great difficulty they had to contend with in preventing deposit was that of keeping the circulation in such a state as that the currents would prevent the deposit, for it was found that where there was a current in a boiler the lime did not deposit to any extent. There was certainly no evidence to show that the lime deposited more on account of the presence of a greater quantity of it. The Americans ran with a pressure of 40 lbs. at sea, and they did not appear to suffer much from deposit. They seemed to overcome the deposit of lime by cleaning the boiler whenever they got into port. He was aware of boilers working at 30 lbs. for six or seven years, and the deposit was not greater with that than at lower pressures. He was quite satisfied that the lime deposited with pure sea water, and with sea water having twice or thrice the usual quantity of salt in it, was the same."

It will be noticed that the central portion of Mr. Elder's remarks convey the idea that "blowing off" *will* cause the desired effect.

Now, while admitting that the theory in question is correct in principle, consideration must be devoted to the gain in practice, not only as to its efficiency, but also the economy. It is not essential in the present day to make any comment on the fact that to discharge a given amount of heat from a boiler—either in the form of water or steam—must cause a proportionate reduction of the relative contents, and that to replace the loss involves an extra consumption of feed water and fuel. Now, when discharging from the surface—or what is commonly known as "surface blow off"—the same amount of waste in the form of water is not occasioned as when using the "blow out," or bottom discharge. The truth of this is strikingly apparent while remembering the relative volumes of the steam and water. The vapour escaping causes a *proportionate* loss of the fluid, but the discharge of the latter is its *actual* removal in cubic contents *without* relation to any other property. It must not be overlooked either that there is a corresponding loss of *heat* by the respective means of discharging, and this involves attention when considering the remedy required. It needs no figures to prove these facts, although in some instances calculations are essential; but in the present case the matter explains itself by natural laws, the validity of which needs no advocacy.

Adverse opinions are of course entertained by the authorities on the present subject, and, therefore, to these a brief notice must be given. Mr. R. Murray, in his work on marine engines, states:—

"The following table shows the boiling point and specific gravity of sea water (at 60° Fahr.) of different degrees of saturation, expressed in parts of salt contained

therein, the barometer indicating 30 inches of mercury.

	Saltness.	Boils.	Sp. gr.	
Pure Water .	0	212°	1·	
Common sea water	$\frac{1}{32}$	213·2°	1·029	
	$\frac{2}{32}$	214·4	1·058	
	$\frac{3}{32}$	215·5	1·087	
Up to this point no deposit will be formed . . .	$\frac{4}{32}$	216·7	1·116	
	$\frac{5}{32}$	217·9	1·145	
	$\frac{6}{32}$	219·1	1·174	
	$\frac{7}{32}$	220·3	1·203	
	$\frac{8}{32}$	221·5	1·232	
	$\frac{9}{32}$	222·7	1·261	
	$\frac{10}{32}$	223·8	1·290	
	$\frac{11}{32}$	225·0	1·319	
	$\frac{12}{32}$	226·1	1·348	saturated solution.

"As a general rule, the atmospheric boiling point of the water should never be allowed to exceed 216°. The temperature must be ascertained by drawing off a small quantity of the brine, and boiling it in a deep copper vessel in the engine room, a correction being made, if necessary, for the state of the barometer.

"The following Table shows the height of the boiling point in Fahrenheit's scale at different heights of the barometer.

Barometer.	Boiling point.	Barometer.	Boiling point.
27 inches.	206·96°	29½ inches.	211·20°
27½	207·84°	30	212°
28	208·69°	30½	212·79°
28½	209·55°	31	213·57°
29	210·38°		

"It will be seen that if we would preserve the water of the boiler at a degree of saturation indicated by $\frac{2}{32}$ of salt, we must blow off one-half of the feed water; if at $\frac{3}{32}$, then one-third must be blown of; at $\frac{4}{32}$, one-fourth, and so on. We have said that $\frac{3}{32}$ is the highest degree of saturation that should be permitted. The degree of saturation of the water should be tested at least once every hour."

It will be seen that Mr. Murray advocates a continual *loss* of one-third of the feed water, to prevent incrustation.

In a paper read by him at the "Institution of Naval Architects," in 1865, he states :—

"A good deal of misapprehension exists as to the loss entailed by the process of blowing off, but I think the following rough calculation will show that it cannot be very great. The feed water enters the boiler at, say 110°, and is blown off at 220°. The fuel required for raising this quantity of water by 110° is thus thrown away. Now the steam which is evaporated had absorbed altogether about 1,210° of heat sensible and latent, and we know that the water blown off bears to the water evaporated a proportion of, say one-half, therefore the heat thrown away in the brine will bear the proportion of one-half of 110 (or 55) to 1,210, which is one twenty-second part. We may therefore infer that 5 per cent. only of the whole quantity of heat utilized in the boiler has been lost by blowing off.

Professor Rankine, when alluding to the properties of sea water, says in his work, to which prior allusion has been made :—

"Ordinary sea water contains about $\frac{1}{32}$ of its weight of salt. The brine in the boiler should never be allowed to rise above *treble* that strength; and for that purpose the volume of brine discharged should be equal to *half the volume of the net feed-water*. But is it

better still to provide that the brine in the boiler shall never rise above *double* the strength of ordinary sea water; and for this purpose the brine discharged should be *equal to the feed-water in volume*."

When treating this subject chemically, he states: "The deposition of carbonate of lime can be prevented by dissolving sal ammoniac in the water; for that salt and the carbonate of lime are mutually decomposed, producing carbonate of ammonia and chloride of calcium, of which both are soluble in water, and the former is volatile. The deposition of sulphate of lime can be prevented by dissolving carbonate of soda in the water; the products being sulphate of soda and carbonate of lime, of which the former is soluble, and the latter falls down in grains, and does not adhere to the boiler."

From these conclusions it is evident that "blowing off" has been considered, by the several authorities quoted, as the most efficient means for the *prevention* of deposit on the internal surfaces of the marine boiler. Apart from this it is certain that the means adopted were put in practice *after* the water was in the boiler. Now a casual observer might coincide with the conclusion that this way of dealing with the matter in question was a reverse mode, inasmuch that the water should be purified *before* it entered the boiler. On reflection this seems feasible, and doubtless the origin of surface condensers.

In the present allusion to this last-mentioned portion of the marine engine, no notice will be taken of its arrangement or detail, but merely the effect of the fluid, after passing through the surface condenser, on marine boilers.

As practical evidence in all matters of science is of respective value, it has been deemed expedient to introduce the following paper by James Jack, Esq., of Liverpool, "On the effect of Surface Condensers on Marine Boilers," read at the Institution of Mechanical Engineers, 1863.

"As Mr. Jack's firm has constructed and fixed a considerable number of surface condensers during the last three or four years, and as certain actions have been found to take place on the tubes and plates of the boilers with these surface condensers, of such a character that the full advantages of the use of distilled water could not with impunity be obtained, it was the purpose of the present paper to give the particulars of these effects. The effects produced upon boilers where surface condensers are used must have been noticed by many engineers, and the object of the paper was therefore to induce discussion upon the subject, and elicit information which will enable the great advantages in the saving of fuel resulting from the employment of surface condensers to be realised. As the boilers where surface condensers are used are insidiously and rapidly acted on, the danger of delay and of accident from explosion is thereby greatly increased, rendering the question one of serious importance.

"Surface condensation is a process by which both the sensible and the latent heat of the steam are conveyed away; and, although the adoption of any new system is necessarily slow, Mr. Jack did not doubt but that surface condensation will ultimately entirely supersede the jet. There are evils however which require remedying before surface condensation can be universally adopted for steam ships; not in the condenser itself, but in the effects produced on the boilers by distilled water or something

contained in it resulting from surface condensation. Mr. Jack accordingly referred first to the effects of surface condensation on the boilers, and secondly to the probable cause of this destructive action.

"A number of marine boilers which came under Mr. Jack's observation, and may for convenience be distinguished by the letter A, had been in use for longer or shorter periods, none less than six months, supplying steam to engines having injection condensers. Salt water had been used for feeding the boilers, and a considerable incrustation had consequently taken place. Without going to the trouble and expense of cleaning these boilers, they were sent to sea immediately after the surface condensers were fitted in. Everything went on satisfactorily, and on returning from the first voyage the boilers were examined. It was found that the greater portion of the incrustation had fallen off, and that the surfaces of the boilers, that is the inner surfaces, were in very good condition for transmitting heat, showing that the adoption of distilled water for feed had been advantageous. Boilers of this class have uniformly had the incrustation nearly all removed by the action arising from the use of the surface condensers during the first voyage, and in every case the surfaces of the boilers have been found in good condition and still remain so, some of them now for as long a term as four years. Indeed, surface condensation has here been in every sense a decided success.

"The appearance of the inside of the boilers, however, was not that of clean iron. The surface seemed to be impregnated with some greyish matter, or to be altered in its chemical nature. That this impregnation or alteration of the surface prevented, and still prevents, injurious action on the metal, will be gathered from a description of another lot of marine boilers, distinguished by the letter B, which were in all cases new boilers, sometimes with new engines and surface condensers, in other instances with old engines and new surface condensers. In port, before starting, a number of these boilers were filled with fresh water, while another number were filled with salt water. An examination after the first voyage, during which only distilled water had been used for feeding the boilers, showed the following effects, which were increased in every subsequent voyage, until the practice was adopted of feeding with say from one-sixth to one-tenth of salt water. First, both above and below the water line, the surfaces of the plates, tubes, and rivets were covered with a deposit resembling hydrated oxide of iron, which when the water was evaporated was in the state of a fine impalpable brownish coloured powder. This deposit was thickest above the water line, sometimes averaging $\frac{3}{4}$ inch thick. When the boilers were emptied a thick slimy deposit adhered all over the inside, an analysis of which showed that it consisted of

Oxide of Iron	77·50
Moisture	19·75
Grease	0·85
Sulphate of Lime	0·80
Oxide of Copper...	0·60
Traces of Alumina and Chloride of Sodium and Magnesium ...	...
Loss	0·50
	100·00

"Secondly, underneath this deposit the plates

and tubes were found to be eaten into, indented, or 'pitted.' The indentations varied in diameter from the smallest speck to $\frac{5}{8}$ inch, and in depth from the merest impression to the entire thickness of the plates or tubes. And although they were formed all over the boilers, they were most frequently found and were most numerous just over the fireplaces, and in those parts immediately in connection with the greatest heat. In some of these parts the surface was entirely covered with the indentations; while in other parts as much as a square foot of plate, although subjected to the greatest heat, was free from them. The plates and tubes in all cases have been of the best iron and by good makers, and the 'pittings' occur in what looks like iron of good quality, with a good fibre, no slag or cinder being perceptible. So destructive was this pitting in boilers, using the same water over and over again, that in one instance the tubes of new boilers were actually eaten through at the end of two or three voyages, extending over only a few months altogether, and it became necessary to put in new tubes, and to use a portion of salt water for feed, to keep up an incrustation, so that the boilers should not be acted upon. If the iron of the boilers had been all of one make, it would naturally have been concluded that the 'pitting' was due to the quality of the iron; but as the iron of different boilers had been obtained from different makers from time to time, the quality of the iron could not be blamed.

"The presence in the boiler of a soft metal, such as copper from the condenser tubes, it was considered would induce a galvanic action such as might affect the iron in some way. But the analysis which was made of the deposit scraped from the boiler shows that there was scarcely a trace of any foreign metal there. Indeed it might have been concluded that a soft metal could not be present, for the tubes of the condenser and the copper pipes were all in a perfect condition. Even at the joints, made tight by india-rubber, hardened by vulcanising, there was scarcely a speck of corrosion.

"A search was then made to ascertain whether the gluey deposit was present that arises from the decomposition of the tallow and oil used for lubrication, as Mr. Jack had frequently heard that such a deposit took place in boilers where Hall's surface condensers were used. For the purpose of ascertaining this, the mud cocks of a vessel were not opened for some time before arriving in port; and the fires were then put out on arrival and the mud discharged, when the only substance found was the watery brownish deposit before referred to. The deposit remaining in the bottom of the boiler was carefully examined, but here again there was only the same deposit. As it was believed that the lubricating material carried into the boilers with the feed might, by continued subjection to heat, form an acid capable of producing the effects observed, the kind of lubricating material employed was noticed, in order to ascertain whether animal or vegetable oils acted most injuriously; but it was found that the action went on as much with the one oil as with the other. In case however a fat acid, formed as already mentioned, might be the cause, pieces of chalk were put into the boilers, and from time to time fresh pieces were added; carbonate of soda was also mixed with the feed water in regular doses; but all to no purpose; the action went on getting worse and worse.

"No other alternative was therefore left, nor is there at present any other as far as Mr. Jack has been able to learn, but to feed the boilers with a portion of salt water sufficient to keep a thin incrustation over the surface of the iron. It was suggested that the deposit was nothing else than rust or oxide of iron, and that it was formed by the chlorine present in the small proportion of salt water, which would combine with the iron to form chloride of iron; and this being readily decomposed by oxygen, oxide of iron would result. The difficulty here, however, was to know whence the oxygen was obtained; for the quantity of air entering with the feed water must have been very small indeed. It was also suggested that hydrochloric acid might be present, from the small quantity of sea water that may have found its way into the boilers; but then the difficulty was to know where a quantity of the acid was to come from, sufficient to act over such an extended surface, and as rapidly as the results showed.

"It was found however by Mr. Rollo, one of Mr. Jack's partners, that in a pair of boilers at a sugar refinery there was the same brownish deposit adhering all over the boilers, and those parts subjected to the greatest heat were 'pitted' in precisely the same manner as the second lot of marine boilers previously designated by the letter B. Exactly the same effects were being produced. These boilers were supplied with the same water over and over again, a small quantity of fresh water being added from time to time to make up for the loss. As the steam was passed only through iron pipes for melting the sugar, the damage to the boilers could not result either from the steam coming in contact with a soft metal, or from any lubricating material. The boilers were of the Cornish construction with one flue, and were worked at about the same pressure as the marine boilers B already referred to, say 20 lbs. per square inch pressure. A pair of boilers of exactly the same construction, placed alongside the first pair, and working at about the same pressure, but fed with water which had not been distilled, were then examined, to learn what state they were in. But although put in about the same time as the two first examined, these boilers were found in good condition and likely to last for years, as there was not a sign of corrosion or 'pitting;' whereas the two boilers working with distilled water had to be repaired.

"The practical knowledge thus acquired necessarily led to the conclusion that the distilled water itself was the cause of the corrosion, instead of any galvanic action or any fatty acid. In reference to the question whether distilled water has any particular action on metals, the chemist Berthier found that nodular protuberances deposited on iron pipes containing distilled water consisted of 21 per cent. of protoxide of iron, 58 per cent. of peroxide of iron, 5 per cent. of carbonic acid, 14½ per cent. of water, and 1⅓ per cent. of silica. The iron pipes contained also a pulverulent substance, which could be produced at pleasure with distilled water to which a trace of carbonate of soda and common salt had been added, but not with an addition of caustic alkali. Distilled water is known to act powerfully on lead, and this action is attributed by Dr. Clark to the remarkable property that distilled water has, as compared with ordinary water, of dissolving free carbonic acid.

"Mr. Jack did not presume to state confidently that distilled water is really itself the active destroyer of iron boilers; but, from the observations that have now been referred to, and the information he has been enabled to obtain, he thinks there is sufficient evidence that distilled water is, if not the sole cause, at least an active agent in producing the corrosive effects that have been described. If this suggestion should lead to the remedy of the evils that have been experienced where distilled water alone has been used, another difficulty will have been overcome towards the complete introduction of 'surface condensers.'"

This paper elicited a discussion usual on these occasions, of which the following is a portion:

Mr. D. Rollo observed "with regard to the boilers at the sugar refinery that had been referred to, the corrosion could not, he thought, have been caused in that case by the presence of brass particles in the water; for the steam passed only between the two cast iron plates of the evaporating pans, and when condensed returned again to the boiler, without coming in contact with any brass at all in the apparatus, the leakage being made up by the addition of a little fresh water. There were four boilers all working together, two fed with the distilled water returned from the evaporating pans and two with fresh water: the two latter at the time of examination were found in perfect condition after four years' work, having only a large deposit or scale of lime over the surface of the iron; while the other two fed with the distilled water, after the same time of working, had become quite unsafe from corrosion and had to be very extensively repaired. They had tried various remedies for the evil: suspending zinc plates in the boilers, but with what result had not yet been ascertained; cast iron pipes instead of copper pipes for conveying the steam; and tinned tubes, zinc, and galvanised iron tubes in the condensers, instead of brass. Block tin tubes had also been tried, but were found not to have strength enough to support their own weight when placed horizontally. All these trials, therefore, still left the question of the cause of corrosion undecided: but he thought it could not be the action of copper, because the analysis that had been made of the deposit found in the boilers using surface condensers showed a very small proportion of copper, the main metallic ingredient being iron, and there was not sufficient copper collected from the boilers in the voyage of a steamship to account for such an extent of corrosion as was met with. Where much grease, however, was used in the engine he had seen the inside of a boiler present an appearance which, he thought, rendered it just possible that the fatty matter or acids contained in the grease had something to do with deteriorating the quality of the water and causing its corrosive action.

He further explained that, in all the vessels they had fitted up with surface condensers, more than twenty-five in number, the boilers had been tubular boilers, all with iron tubes, excepting only one or two cases where brass boiler tubes were used; and in all cases the condenser tubes were brass. With regard to steel boiler tubes, he had seen them tried, and with no better results than the iron tubes; indeed, in some instances, they became corroded rather quicker than iron tubes, but whether that was owing to the additional carbon contained in the steel he could not say.

With reference to "priming," he stated: "he was satisfied that priming had nothing to do with the use of surface condensers; because in no case had they had to make any alteration in the boilers after applying the surface condensers, nor had the priming ever been found to be worse after the surface condensers were applied than it had before been with the jet condensers previously used with the same boilers. If the boilers had plenty of steam room, he had found there was not any danger of priming; but if that point were neglected and the roof of the boiler was low, water would get carried over into the cylinders along with the steam. In support of the theory that the water underwent some change by the repeated boiling, where surface condensers were used, he understood it was found that fresh water after continued use in that way carried the salinometer higher than salt water, showing that it had then become denser than the salt water used in marine boilers was generally allowed to get before changing."

Mr. William Clay remarked "that there seemed to be two theories suggested to account for the corrosion experienced in boilers working with surface condensers: first, that a certain galvanic action was produced by particles of brass or copper carried into the boiler by the water from the condenser tubes, whereby the plates were corroded wherever a particle of those metals was deposited; and secondly, that acids produced from the grease brought over from the engine might have the deleterious effects that were noticed. In favour of the former theory, that the galvanic action occasioned by brass or copper caused the injury, was the peculiar shape which the corrosion assumed: it was not shown by a regular thinning of the plates, but they were pitted in particular spots, whilst the iron in the immediate neighbourhood was as thick and as good as ever; whereas, if some acid had been acting on the plates, it would be expected that the action must have been more uniform, and extended over the whole surface, instead of being concentrated at particular points. If the use of brass or copper tubes in the surface condensers had been the cause of corrosion, he suggested that the plan of Mr. Lamb's boilers, with a number of flat flues of iron plates exposing a large extent of surface, might be efficient for surface condensers also, to avoid having any brass or copper at all in their construction. He had himself experienced corrosive action in a number of boilers which had no brass work about them, except the steam valves and valve boxes, and on certain portions of the iron plates a corrosion took place exactly like what had been experienced in the cases already described. One boiler in particular was attacked out of a set of 25, and for a considerable time the corrosion could not be accounted for. The corroded iron plates were removed and steel plates substituted for them, but these also were eaten away and pitted exactly in the same manner. Then the water was changed, and fresh water was used from the town supply or from a well; this was constantly changed, so as not to use the same water over and over again, and new steel plates were put in to replace the corroded ones. In this case no amount of acid could be formed from grease, because there was no grease that could get near the boiler, except a little oil to keep the donkey pumps lubricated. The remedy was then found to lie in altering the manner in which the feed water was injected.

The boiler was a vertical one, 30 feet high, with two large flues through it, and the feed water was originally injected at the bottom, impinging upon the flue; and the pitting of the plates took place for a length of 3 feet from the entrance of the feed. But by altering the feed pipe to enter at the centre of the boiler and furnishing it with a rose jet to distribute the feed water gradually, the corrosion was completely stopped at once." He enquired whether it had been found that there was any particular way of supplying the feed water in the boilers described in the paper, in order to diminish the corrosive effects.

Mr. D. Rollo replied, that "he had not found that the place where the feed was introduced in the boilers had any perceptible connection with the corrosion: the feed pipes had been altered to different positions in several of the boilers described in the paper, and in some had remained unaltered, but without affecting the corrosion; and the corrosive action seemed to have no special effect at the parts nearest the entrance of the feed, but the pitting seized upon one plate and another indiscriminately and with considerable spaces between. Priming no doubt was effected by the position of the feed entrance; and in one of the vessels that they had fitted with the surface condensers, having round tubular boilers, much trouble had been experienced from priming, in consequence of the feed being injected too near the point where the steam was taken off; but by putting the feed pipe lower and at the back of the boiler, the priming was now got rid of completely. He did not think the corrosion of the boiler plates was caused by grease contained in water: but it seemed more probable that, by constantly boiling the same water over and over again, it was robbed of some of its original properties, or became otherwise altered in quality thereby, so as to produce the serious effects that were experienced.

Since this evidence, from the various authorities now quoted, much has been done in the way of research and reformation, and the result has been successful. Before, however, entering into this subject, a further notice will be given to the "action of the distilled water on the boiler plate."

Surface condensers are for the most part composed of brass tubes in the present day, this, as much for practical purposes as theoretical. Brass tubes can be drawn out without seam, also the action of the fluid on them does not displace much of the entire nature of the material. Now a copper tube—although the better conductor of heat and cold—is more expensive in manufacture and use than brass, and *does* impart more of its nature to the fluid during the passage of the same than the former material. Practical evidence is often seen as to the truth of this, that where copper and wrought iron are in contact a galvanic action ensues, and the latter material succumbs to the power of the former. It is obvious then, that the condensing tube—if of brass even—must be robbed of some of its copper properties during the act of condensation, and the fluid is impregnated with the same. On the water entering the boiler a galvanic action ensues, similar in principle to that produced by the actual contact of copper and iron.

Next must be noticed the form assumed

by the corroded parts. It is found in practice, as already alluded to, that the surface of the plate presents minute hollows, of different shapes, and in some cases separate from each other, therefore, correctly denominated "pitting." Now while observing that this pitting is far from what might have been supposed to be the result of galvanic action, viz., a gradual wearing or thinning of the plate throughout, it must be remembered that the material in question is not as homogeneous in its property as the copper, and this doubtless is one of the causes for the uneven corrosion. Again, for example : the properties with which the water is impregnated may not combine under a given temperature, forming thereby a separation of metallic matter in the fluid, and thus another cause added to that already alluded to.

Apart from the metallic agent, another has been proved to be equally destructive—the lubricant used in the cylinders. Not only is it corrosive, but it also forms a sediment both in the condenser and boiler, and therefore also a chemical action is doubtless the result of contact; which has been proved, by the grease forming an acid of great consuming power on metallic surfaces.

It will thus be seen that, if the marine engineer gained one desired effect by the introduction of surface condensers, he introduced others of an evil nature, almost sufficient for him to regret the production of his powers of thought. It may be stated that the corrosion of the boiler plates has ensued without the feed water coming in contact with copper or brass surfaces; this is accounted for by the fact that the water contained some mineral property, and thus a galvanic action ensued. In other instances, the oil or tallow used in the cylinder caused a chemical action, and thereby the same result as to corrosion.

Thus far having treated of the baneful effects of surface condensation on marine boilers, attention must now be given to the best means yet practised to counteract or prevent the same.

It will be noticed that entire sea water as feed water produces certain incrustation; it has also been stated that distilled water obviates the same. Now the latter destroys or corrodes the surface of the plates in actual contact with it. It is, therefore, obvious that, to prevent corrosion by galvanic or chemical action, an interception must be formed between the fluid and the plate, and be a relative non-conductor at the same time. Here also is the fact that a sediment is requisite, but the main question is; by what means, and the required thickness to admit the penetration of the caloric from the furnace? Practice has, however, some time ago settled this matter, of which we have had some experience, the solution of the problem being thus effected. The steam when in the cylinder has not been permitted to be impregnated with any acid—therefore the use of tallow or grease for internal lubrication has been abolished. Also, it has been proved, the incrustation formed by entire sea water is not injurious to the boiler, by chemical or galvanic action, and the lodgment of sediment of the required thinness is thereby attained — by the introduction of a small quantity of sea water in the distilled-water tank, before feeding the boiler—and thus a slight incrustation is formed, not injurious as a non-conductor of heat, but

efficient as a non-conductor of corrosive action.

Now as to the quantity of the sea water to be admitted, practice is the best teacher; and selecting, therefore, one example of certain authenticity amongst others, the following is the result. In a steam vessel fitted with engines of 350 horse-power—nominal,—collectively—surface condensation and superheated steam being adopted—sixteen to twenty gallons of sea water per watch of four hours was introduced in the tank, and the boiler was thereby scaled about a thickness of one-sixteenth of an inch only in three months. This sediment was removed at intervals by blowing out, when in port after the fires were extinguished; in a few hours the scale cracked, and a brush only required to cause its effective removal; also the plates showed no indications of corrosion or wear. This mode of treating a boiler having been proved to be efficacious after some years' trial, it remains but for others to adopt the same with an equal result.

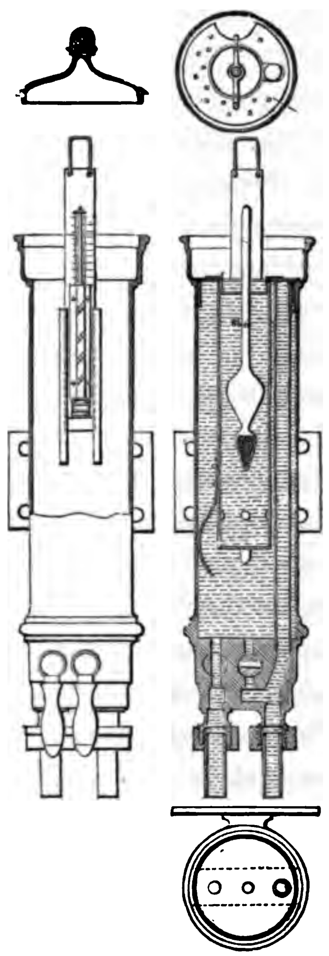
USE AND EFFECT OF SALINOMETERS.

The salinometer bears such a near relation to the marine boiler in the present day, and its adoption being in strict allusion to incrustations, it has been thought expedient to add a notice of the use, and description of the same, as a conclusion to the present chapter. In rendering its practical utility obvious, the system of condensation will not be considered; as the instrument in question is essential in either case.

Until the introduction of salinometers, the process of testing the density of the water in the boilers was accomplished by blowing out or off, and this was done at intervals, determined by the random judgment of the person in charge. It is obvious, however, that if this operation is performed too often, water may be blown out which has not yet reached the degree of saltness requiring the same. The consequent introduction of more cold sea water into the boiler will cause increased consumption of fuel to restore the lost heat, and it is estimated that from this cause the firemen are compelled to feed the boiler at least one-fourth oftener. On the other hand, should the operation of blowing out not be performed at the proper periods, incrustation may take place to a highly injurious degree in a very short space of time. It will, therefore, be at once apparent, that it is of the highest importance that the engineer should be able at all times to ascertain with precision the density of the water in his boiler. In order to do this, an adaptation of the hydrometer, graduated for the special use to which it was to be applied, was made, and soon obtained favour with marine engineers. As, however, every time it was necessary to use this instrument, water had to be drawn into a can or bucket, and as in rough weather this was a feat almost impossible of achievement, it soon occurred to Mr. Sewell, a practical American engineer, that for the proper and constant use of this instrument—more especially its adoption in connection with the thermometer, without which its indications are not thoroughly reliable—it should be provided with a fixed *case* in which it could float freely, and in which the hydrometer's indications, together with those of the thermometer, might at all times be seen; also that the combined instruments and their case, forming altogether the "salinometer," should be rendered a continuous *indicator* by

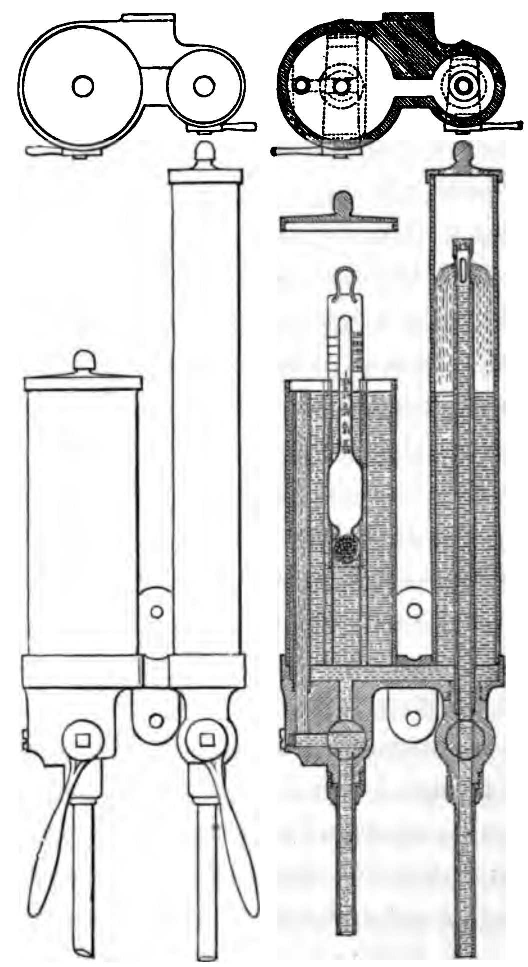
maintaining a certain stream of water through the case. The result of these reflections was the production of Sewell's, or, as known in this country, "How's Salinometer." This instrument is one of the best arrangements, being available as a continuous indicator, with the modifications and improvements of Mr. Gathercole, the successor to Mr. A. P. How. It will be seen by referring to the above illustration, Fig. 37, that this instrument consists of a single cylinder connected with the boiler by a pipe which is provided with a stop cock. Water is admitted to the salinometer by the left-hand cock in illustration, it flows thence around and into the hydrometer tube, and then through the waste pipe as shown. One of the features of this improved instrument, is that the inlet and outlet are so proportioned that all water admitted must escape without overflowing or scalding the operator. The safety of the hydrometer is also ensured by providing for it a tube within which it may float tranquilly, so that any steam admitted into the salinometer will be disengaged before the water passes into the hydrometer tube. In the early salinometers of "Sewell and How," this tube was not provided,

Fig. 37.



HOW'S PATENT SALINOMETER.

Fig. 38.



LONG'S PATENT SALINOMETER.

but it soon became apparent that attention should be paid to the proper relative proportion of the inlet and outlet. It has happened in many cases, that where too great a stream of water is maintained, causing oscillation of the hydrometer, or where water is admitted only occasionally, as it is required from time to time to ascertain its density, a rush of steam has sometimes driven out the water and scalded the operator, and, in some cases, broken the hydrometer. To provide against such occurrences, Mr. Long, an engineer of the United States Navy, modified the "How or Sewell" instrument, in the manner represented in the illustrations, Fig. 38 on page 190. To the "How or Sewell" cylinder of the early construction another cylinder is added with an internal tube. The water from the boiler is let into this internal tube, and escaping at the top, falls down into the additional cylinder. In doing this the steam becomes disengaged, and the solid water passes into the "How or Sewell" cylinder, in which the hydrometer and thermometer are placed.

The advantages which result from the proper use of salinometers are now admitted to be beyond dispute, and in some form they are to be found in every sea-going steamer, the results of their judicious employment being certain economy of fuel and the prevention of injury to the boiler. The neglect of proper attention to the state of marine steam boilers cannot but be followed by great loss, and, therefore, each boiler should be fitted with a salinometer containing the indicating instruments — an hydrometer and thermometer—so placed as to admit of instant and constant inspection. Through the salinometer also a stream of water should be continuously maintained. In addition, the instrument should be provided with means for the purpose of regulating the stream, to maintain the requisite degree of temperature; also, attention should be given to the proper location of the instrument and the connection between it and the boiler. Moreover, if it be of importance, as it really is, that the tubing connecting with the boiler be of a sufficient diameter to ensure a constant supply of water in case incrustation or accumulation of salt should take place, and that this tubing should have no unnecessary sharp bends and contractions, it is manifest also of no less importance that the salinometer itself should not have contracted or tortuous passages, and that the inlet, though capable of being regulated, should be of such capacity of admission as to prevent being easily choked.

The following explanation will render the operation of the salinometer easily understood.

It is estimated that 1lb. of salt enters the boiler with every 32lbs. of sea water, so that when one-half, or 16lbs. out of 32lbs. are evaporated, the 1lb. of salt is in the remaining 16lbs. of water, and is then in the proportion of 2lbs. of salt to 32lbs. of water. Then, if one-half of the remaining 16lbs. of water be evaporated, there will be 8lbs. left, in which will be the 1lb. of salt that entered with the 32lbs. of water; it would then be in the proportion of 4lbs. of salt to 32lbs. of water, or, what is called $\frac{4}{32}$ on the hydrometer. Now, if this 8lbs. of water were blown out, all the salt that entered in the 32lbs. of water would be discharged also; and, as long as this proportional part of the water that entered was blown out, the water in the

boiler cannot attain a greater degree of saltness than $\frac{4}{32}$; from which it will be seen that, to keep the water in the boiler at that density, one-quarter of the water that enters must be blown out; and to keep it at $\frac{2}{32}$, one-half must be blown out, and in a similar proper proportion for any other density when using sea-water.

The hydrometer generally used is graduated, marked upon the principle just explained, and is fig. 0 for fresh water—$\frac{1}{32}$ for sea water, which contains 1lb. of salt to 32lbs. of water—$\frac{2}{32}$ when there is 2lbs. of salt to 32lbs. of water—$\frac{3}{32}$ when there is 3lbs. of salt to 32lbs. of water—and $\frac{4}{32}$ when there is 4lbs. of salt to 32lbs. of water, and so on. Each division is subdivided into four parts, showing halves and quarters of each.

The hydrometer is also graduated for the temperature of 200 degrees, it being necessary to have some standard of temperature at which the indications are always to be taken; as steam of different pressures has different temperatures, and as a difference in temperature will alter the indications of the hydrometer, it is, therefore, necessary that the water be relieved from the pressure in the boiler, that it may assume a uniform temperature. This it will do under the pressure of the atmosphere, and it is thus accomplished in this instrument, which renders it applicable to boilers in which the pressure of the steam may vary, or in which either high or low pressure steam is used.

The "Gathercole-How" and the "Long" salinometers are manufactured by Messrs. Soul and Co., of Finsbury-square, who kindly supplied the working drawings from which these presented were reduced.

Enough has been stated to render obvious the defects common to the use of condensation with marine engines. It will be observed, on reflection, that, although the density of the *sea* water has been quoted, practically the boiler—even with injection condensers—is fed from the tank or discharge pipe, as a rule; and it is only when the donkey or sea-feed pump is used that the *natural* sea water is admitted into the boiler.

The gain with surface condensers is only certain when the sea water is partially introduced in the distilled-water tank; and in no case whatever should this be omitted—*i. e.* a perfect amalgamation of the distilled and sea water should ensue before either enters the boiler. There is a just cause for this, from the fact that were the sea water used at intervals in its natural state, an uneven incrustation must result therefrom; and it is certain, also, that an entire amalgamation cannot ensue in the boiler when the requisite evaporation is being effected.

Now, with reference to the regulation of the density of the water in the boiler, and thereby determining the amount of natural sea water to be admitted in the tank, the more efficient means is to allow a continual surface exit from the boiler, generally termed scumming or surface blow off; by permitting also an easy discharge through the salinometer, a correct knowledge of the density of the fluid contents of the boiler is ensured. An efficient practice is to preserve a density $\frac{1 \cdot 5}{32}$ to $\frac{2}{32}$ with surface condensers, and $\frac{2}{32}$ to $\frac{2 \cdot 5}{32}$ with the injection system. As a conclusion to this chapter, it can truly be said that theory based on correct principles must produce practical results.

CHAPTER V.

DETAILS OF OSCILLATING ENGINES FOR PADDLE-WHEEL PROPULSION.

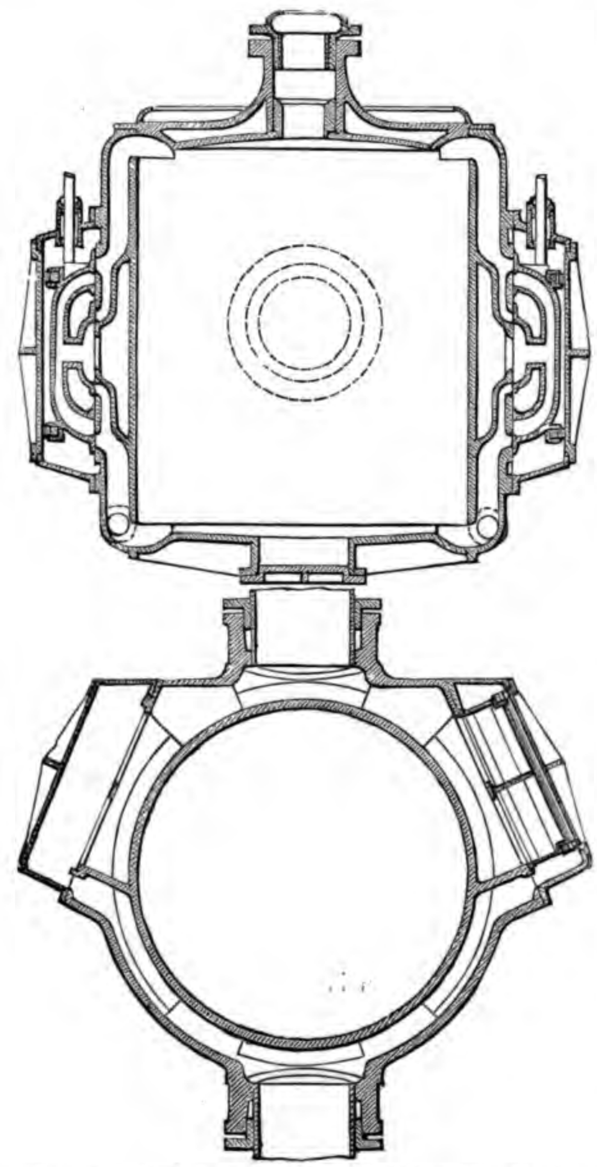
THE details of marine engines form a wide subject for discussion, inasmuch that, although one effect is sought after by the several makers, each differs in design and proportion. In dealing with this matter, practical examples will be quoted and dimensions adverted to, thus forming a certain standard for productions of greater or less power. The illustrations, relating to the engines under direct notice, are composed, for the most part, of details from the examples represented by Plate 30, produced by Messrs. James Watt and Co.; but where any deviation is made as a comparison, attention to the same will be directed, but unless this occurs, it will be remembered to whose engines the details belong.

CYLINDERS.

The cylinders for oscillating engines do not differ much in design, excepting the arrangement of the steam ports and the position of the trunnions. Those latter portions are generally situated at a given distance from the top or bottom end, to retain an equal strain on the piston rod when the cylinder is at the requisite angles. The centre of gravity can also be maintained by proportionating the position of the slide valve chest, and allowing the trunnions to be in the centre of the cylinder's length. The great depth required for the main stuffing box on the cover of the cylinder renders the cover heavy, and thus, if the trunnions and slide valves were at the centre of the cylinder's length, the upper portion would be heavier than the lower, instead of both equal, as required. In some cases it is preferred to make the centre of the trunnions, and that of the exhaust ports, on the same horizontal line, rather than as shown by the sectional elevation, Fig. 39—page 194. In this example it is seen that the trunnions are raised, while the valve casings are lowered from the centre of the cylinder's length; or, to be still more explanatory, the top of the exhaust passage, and that of the trunnion tubes, are on the same level; but the belt surrounding the cylinder is equidistant, vertically on each side of the trunnions' centres. The cylinder cover and stuffing box are of the ordinary kind, with a shallow gland and deep bottom bush. The raised curved portion on the cover serves alike for ornament and a repository for any waste lubricant. The bottom end is cast with the body, and the boring bar can be passed through the hole provided. The slide valve is double-ported, of the equilibrium type, packed at the back with india-rubber and a face ring. The sectional plan renders a ready conception of the arrangement of the supply and exhaust belts, or steam passages, and the angular position of the slide valves and casings. The trunnions, stuffing boxes, glands, and tubes, are the usual mode,

both as to design and proportion. The relief valves openings are shown, with the facings in dotted lines in the sectional elevation, at the bottom steam passages. To enable a just appreciation of the merits of the details under notice,

Fig. 39.



MESSRS. WATT'S OSCILLATING CYLINDER AND VALVES.

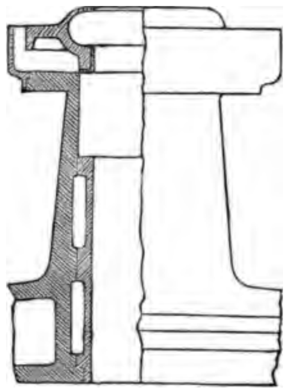
the following dimensions of the same are quoted from the working drawings, from which those herein alluded to were reduced :—

	Ft.	In.
Diameter of Cylinder	0	64
Length of Stroke	5	0
Diameter of Cylinder Cover	6	0
Thickness of Cylinder Body	0	1½
Vertical Distance of centre of Trunnion from the base of the bored portion of the Cylinder	3	2
Vertical distance between centre of Trunnion and centre of Exhaust port	0	6½
Total depth of Belt	3	0
Thickness of Belt	0	1¼
Width of Belt passage	0	4
Diameter of Trunnion bearing	2	4
Length of ditto	0	8¾
Internal diameter of Trunnion tubing	0	17½
Thickness of metal of Cylinder Cover and Steam Passages	0	1¼
Diameter of Piston Rod	0	9
Diameter of Stuffing Box	0	12
Depth of Stuffing Box	0	13
Depth of Bottom Bush	0	7
Depth of Gland	0	6
Stroke of Slide Valve	0	7
Outside Lap do.	0	2⅛
Inside Lap do.	0	0¼
Lead do.	0	0⅛
Travel of the Piston during the admission of the steam by the slide valve, up-stroke	0	40⅝
Ditto, down-stroke	0	32⅞
Width of Supply Ports in Cylinder	0	2
Width of Exhaust Ports in Cylinder	0	6
Width of Large Bars in Cylinder	0	8
Width of Small Bars in Cylinder	0	2½
Width of Supply Opening caused by the Valve	0	1⅜
Length of Steam Ports	2	6
Diameter of Slide Valve Rod	0	1¾
Diameter of Relief Openings in Cylinder	0	3½

The depth of the bottom bush in the stuffing box for the piston rod being of importance, another example of the same is represented by Fig. 40—page 195—shown half in section and half complete. In this case, the lower bush is much deeper in proportion to the diameter of the rod than the other, although the diameter of the cylinder is 61 inches, and the stroke only 4 feet 6 inches. The diameter of the piston rod is 7¼ inches, and the stuffing box

9¾ inches. The depth of the bottom bush is 17½ inches, and the stuffing box 6½ inches, and the adjustable depth for the gland only

Fig. 40.

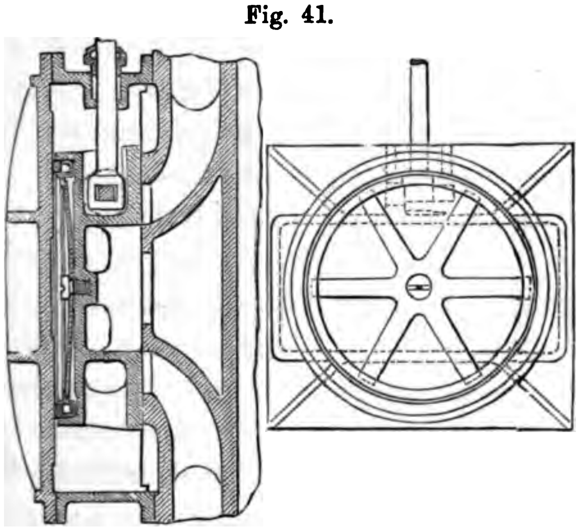


ORDINARY STUFFING BOX AND GLAND OF OSCILLATING CYLINDER.

1½ inch. This example is nearly similar to the practice of Messrs. Ravenhill and Hodgson, and many other makers of paddle engines.

Making further reference to slide valves, the illustration, Fig. 41, is introduced as an

Fig. 41.



ORDINARY SLIDE VALVE FOR OSCILLATING CYLINDER.

ordinary example of a slide valve by many firms, being a single-ported type with a circular facering, and a six-bladed spring under the same, to ensure a given contact or pressure against the facing on the cover. It is seen that a recess is formed on the body of the valve of a diameter nearly equal to the length of the same, and thus the action of the steam on the body is neutralized to a great extent; the surface acted on being only at the corners, and that beyond the periphery of the face ring at the sides.

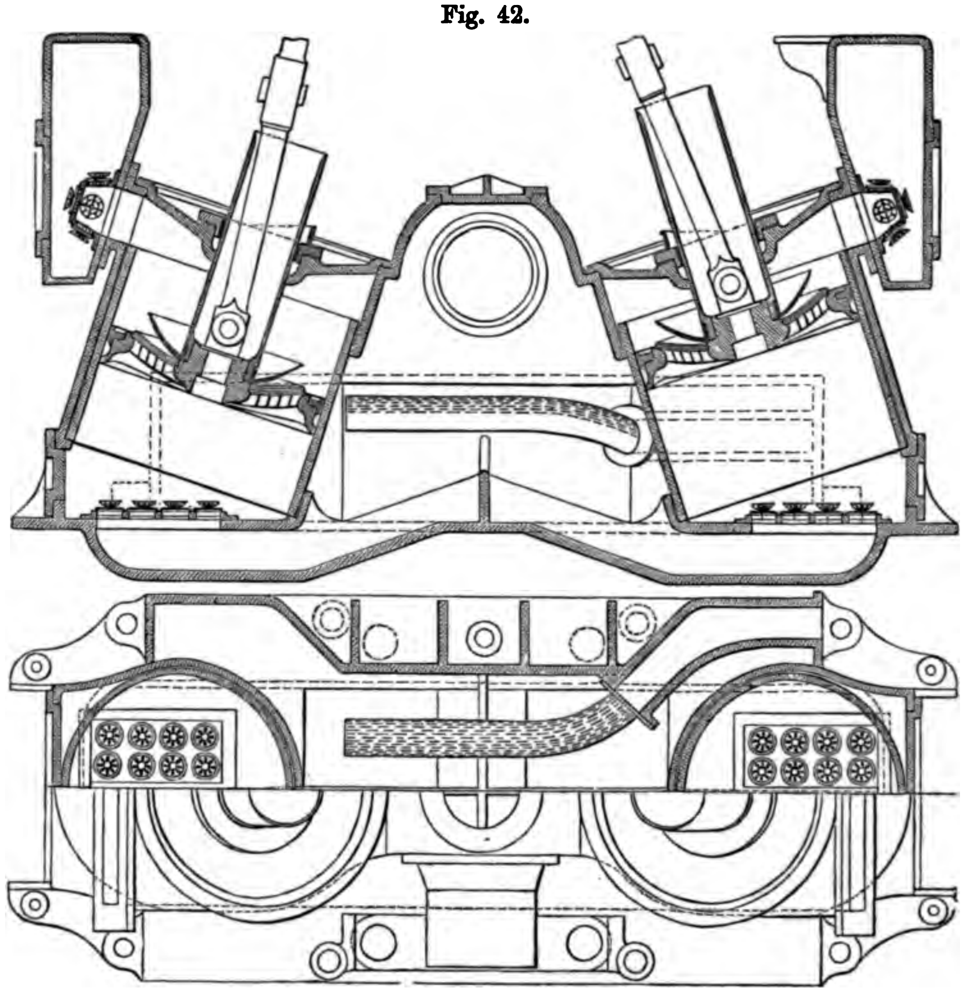
The principle is, therefore, the same as that with Fig. 39—page 194—where the packing is simply of india-rubber under the face ring. Messrs. Watt and Co. have lately preferred to insert the packing and ring in a recess cast with the cover, and the facing is formed on the back of the slide. By the introduction of two rings—the packing between them—a perfect adjustment can be attained by studs on the casing cover. This plan is a favourite mode with Messrs. R. Napier and Sons, in Glasgow, and some engineers in London also adopt it, amongst which may be mentioned Messrs. J. and G. Rennie.

CONDENSERS.

The illustration, Fig. 42—page 196—is the plan and elevation of the condenser and air pumps, in connection with the cylinder already noticed. The pumps are the trunk type with valvular pistons. The foot or suction valves are small in comparison to that on the piston, and the same remark applies to the discharge or delivery valves above the pumps, in the discharge chamber. It will be noticed that the arrangement of these latter valves are peculiar in relation to the others. Instead of a flat seating or a uniform position for the valves, three distinct positions are preferred—vertical, lateral, and inverted.

This novel arrangement of the valves is for one particular attainment, viz., a certain compactness of space occupied transversely and vertically, and by no other position for the valves could this be accomplished. The general practice being either a lateral position, or similarly as the foot valves are, it is obvious then that, with the arrangement in question, room is economised by the adoption of the same. Doors are secured opposite each set of valves for the purpose of inspection or renewal. This acquisition in relation to the foot valves is too often omitted in some arrangements, even by the leading engineers of the day. It is too common the practice to *box* in the foot valves, and depend alone on the removal of the air pump cover, trunk, and piston, to inspect the valves in question only, which result is always prior to repair or renewal.

Fig. 42.



MESSRS. WATT'S INJECTION CONDENSER FOR OSCILLATING ENGINES.

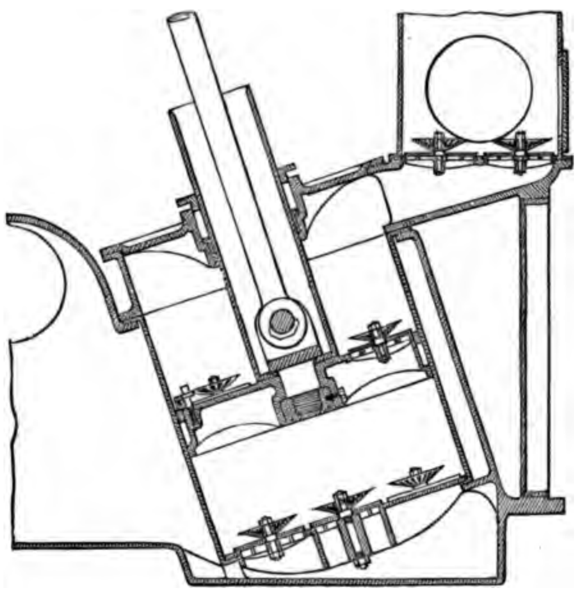
The injection pipe is seen both in elevation and plan, perforated on the upper side only: its position in the condenser is correct in principle and thus certain in practice, being at such a convenient distance from the

trunnion opening, that the termination of the spray is in direct contact with the steam. This matter should always meet with due attention, as the dispersion of the fluid in minute portions will condense the steam more effectually than when a solid bulk of the liquid is brought in contact with the vapour. Two injection pipes are for this purpose adopted in the present case, each being arranged so to divide the allotted duty required with equal action. With reference to the large, or single, circular valve on the piston, many authorities object to its adoption, and ourselves also in common with others. On alluding to this fact to Messrs. Watt and Co., they informed us that, this had been their practice for some years to the present time, and the result had been sufficient for them to warrant its further adoption. Now, while admitting that a larger area can be secured with the single or large valve, it seems, on reflection, to be feasible that valves of a lesser diameter will last longer, and are likewise more portable; also that, although the adoption of the small valves causes more detail, they are undoubtedly more effective with high speeds. The guard of the valve in question is perforated with as many holes as the surface will admit; this, in order to produce lightness of weight, and the certainty of the valve returning to its seat. The following are the dimensions of the principal portions of this arrangement, in connection with those already quoted relative to the cylinder:—

	Ft.	In.
Diameter of Connecting Rod	0	4½
Diameter of Air Pump	3	4
Diameter of Trunk	0	13
Length of Stroke	2	6
Total area of Foot Valves (one pump)	338 sq. in.	
Total area of Piston Valve (one pump)	351·68 sq. in	
Total area of Discharge Valves (one pump)	378 sq. in.	
Diameter of Injection pipe	0	6

The practice with many eminent firms, such as Messrs. J. Penn and Son, Messrs. Maudslay, Sons, and Field, Messrs. J. and G. Rennie, Messrs. Ravenhill and Hodgson, and others of importance, is, to arrange the condenser—half only shown—and air pump, as represented by Fig. 43, in sectional elevation. It is seen that the foot valves are enclosed under the piston, without any means of access—except the removal of that detail. The piston valves are circular, of portable diameters, and arranged around the trunk. The discharge valves are secured beyond and above the pump, on a projection suitably provided for that purpose. It is sometimes preferred by the several authorities to

Fig. 43.



ORDINARY CONDENSER AND ARRANGEMENT OF AIR PUMP FITTINGS.

suspend these latter valves laterally in the exhaust passage, rather than vertically as indicated. By such an arrangement of the details in question, less space can be occupied to produce the required effect than when situated as shown; and the door for access, instead of being above the valves as represented, would be opposite the same. Another cause for laterally suspending the valves is, that the closer these details are to the air pump the more effectual is the discharge from the same. This is a matter of recognised importance with those who understand the subject under notice.

For a pair of air pumps and one condenser, for oscillating engines of 250 horse power nominal collectively, the practice has lately been as follows relative to the proportions:—

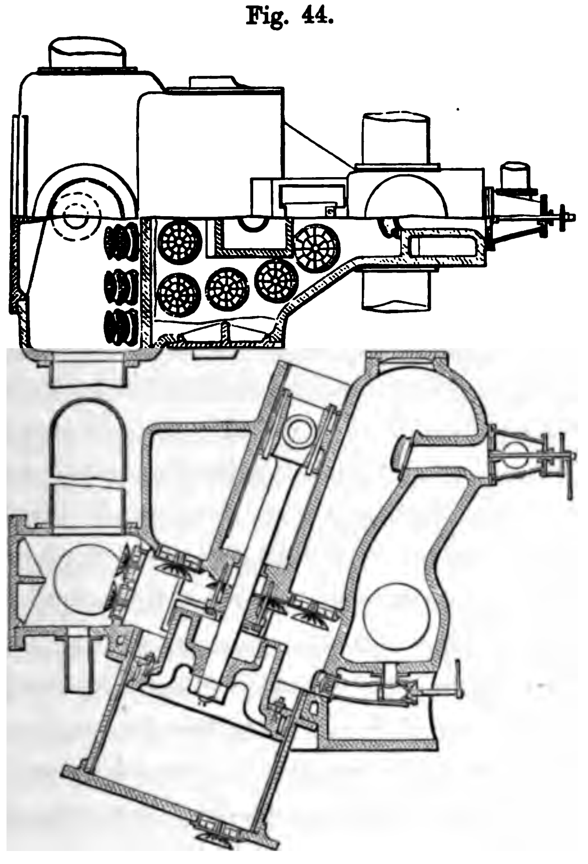
	Ft.	In.
Diameter of Connecting Rod	0	3
Diameter of Air Pump	2	10
Diameter of Trunk	0	10½
Length of Stroke	1	9
Area of Foot Valves (7 for 1 pump) .	276 sq. in.	
Area of Piston Valves (7 for 1 pump)	241·5 sq. in.	
Area of Discharge Valve (8 for 1 pump)	276 sq. in.	
Diameter of Injection Pipe (2) . . .	0	5

In some instances, Messrs. Penn and Messrs. Maudslay have put the discharge valve directly over the air pump piston, allowing the trunk to pass through the top valve seating, and studs or bolts passing through the top cover retain the seating in question in its position. This, although objectionable as an arrangement theoretically, is not found practically inconvenient, as the entire portion above the foot valves with small engines, below fifty horse power nominal collectively, can be removed without a separate disconnection.

Notwithstanding these facts relating to small types, and that doors are arranged to admit access to the foot valves in larger examples, it is obvious that in either case an independent means of access to the separate portions is not available. Another question also presents itself for due attention—the position of the condenser relative to the air pump. It is seen that the general practice, from the two examples previously noticed, is to arrange the condenser and pump on the same level in principle, and thus the water will not rise in the pump—by gravitation—above that in the condenser. The drainage of the condenser is only admitted by the rising of the air pump piston, and should the latter not be in perfect order, a great loss of vacuum is the result. The main cause for the universal adoption of the arrangements in question, combined with the knowledge of their several defects, is that the motion imparted to the pistons is from the crank shaft, and thus an almost vertical action imperative. The condenser also, with large engines of great weight, must, or should be, as low in the hull of the vessel as practicable. The pumps are, as a rule, with little exception, single acting, and thus the foot valves are at the bottom of the condenser, at the base of the pump; the recessed portion, directly below the barrel, admitting an accumulation of the condensed steam and water under the valves in question. On the piston rising, the foot valves open; the water leaves the condenser and ascends in the pump, although the natural course of the fluid is to descend. The piston, when descending, closes the foot valves by causing a load on them, due to the water in the pump; the weight is next relieved from the valves by the rising of the piston, and the final discharge effected.

Now the action of the piston and valves as arranged in the examples alluded to being obvious by noticing the illustrations, the description just presented may seem unnecessary. It must be added, however, as a cause, that although the remarks in question are in connection with the prior examples, they refer comparatively to the arrangement next to be

Fig. 44.



BURGH'S IMPROVED INJECTION CONDENSER AND AIR PUMP.

alluded to, represented by Fig. 44. In this example the condenser is above the pump, the bottom of the former being nearly level with the exhaust opening in the trunnion. The suction valves are inverted, and the discharge valves laterally situated directly below the former. The piston is of a peculiar section, recessed to suit the requirements of adjustment, and effecting an almost perfect discharge from the space directly below the condenser. The piston rod passes through the bottom of the condenser, in which latter compartment a guide channel is formed. The discharge chamber and air vessel are of the ordinary kind, with the feed water pipe shown connected to the underside. The injection valve passage and rose-pipe are near the roof of the condenser, a certain height above the trunnion openings. The cock and valve below the trunnion openings, are for the purpose of permitting a discharge of any condensed steam that may accumulate in the exhaust steam passage at the front of the condenser.

The sectional elevation having been thus far alluded to, attention must now be directed to the plan. To render this view practical as well as conclusive, one half is shown in section and the other complete.

This form of condenser is especially adapted to admit ample space for adjusting and renewing the packings of the exhaust trunnion stuffing boxes, an advantage only to be appreciated by those who are acquainted with the cramped space generally allotted for those purposes. Access to the entire set of valves is attained by doors suitably placed—the representation of which is seen in both views.

Next to be considered is the actual gain by the adoption of the arrangement under present notice over those previously alluded to as the common practice. It will be remembered it has been stated that, with the usual arrangements, on the piston rising the foot valves open. Now in the present

case matters are reversed, the valves, common to the condenser, open only to drain the same, when the piston is descending; after this is accomplished the water is discharged without passing through the piston; thus finally the same result ensues, the water is discharged when the piston is ascending—as before—but the foot valves are closed. Further than this as an actual gain, the gravity of the water assists the action of the pump when draining the condenser. By this arrangement a better vacuum is certain, and one entire set of piston valves dispensed with. Economy of material is thus produced, simplicity of arrangement effected, and the main attainment—perfect condensation—rendered certain. This arrangement under direct notice is originated by ourselves.

For a pair of oscillating engines of 150 horse power nominal collectively, having one air pump, the following are the dimensions :—

	Ft. In.
Diameter of Connecting and Piston Rods	0 4
Diameter of Air Pump	2 10
Length of Stroke	2 0
Area of Suction Valves (10)	260 sq. in.
Area of Discharge Valves (12)	280 sq. in.
Diameter of Injection Pipe	0 4⅝

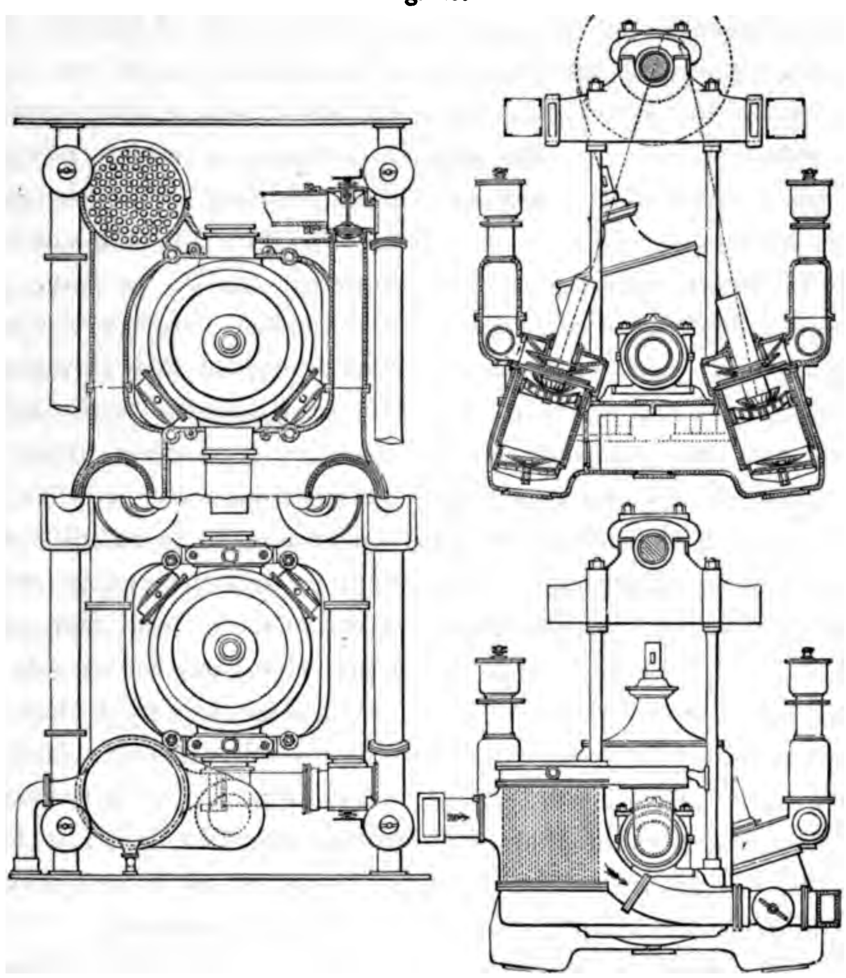
Thus far the injection condenser has been described and illustrated according to the general practice of the present day. It becomes next necessary, before closing the present section, in doing justice to the details under notice, to allude to the surface condensers as applied to oscillating engines adapted for paddle propulsion. The system of condensation now adverted to is not as universal as the injection kind; due, perhaps, more to the numerous details requisite, and the motion required for the pumps, than any other cause, as the latter acquisition is in fact the main attainment to be overcome. With surface condensers two classes of pumps are requisite, one for the exhaustion of the condensed steam, and the other for the circulation of the condensing water. Now, with oscillating engines, two means of imparting motion to these pumps are only available without auxiliary power; either from the vibratory action of the cylinder, or from the rotative movement of the crank shaft. Having thus decided the means available for the given purposes, it will be well next to consider the practicability in either case, and decide as to the better mode. The arrangement of the condenser at present need not be attended to, but immediate allusion being given to the pumps. Now if the motion is imparted direct from the intermediate shaft—whether by crank or eccentric—the pumps will have a vertical action in principle, although they may be situated angularly.

By the introduction of levers, a horizontal movement for the pistons can be produced from the crank shaft. If this latter action is required to be directly derived, the lower end of the cylinder will impart the motion, or by a projecting arm from the upper portion of the cylinder, a vertical action can be given to the pump. It will thus be understood that various means of connection are available, each having a distinct relation to the working parts of the engines. Now, if a vertical action is resorted to, the pumps and valves must be beyond the condenser at the front or sides, thus occupying

a certain space in excess of the outline of the condenser itself; but if a horizontal action is maintained, the pumps and their appendages can be located under the base of the condenser if required.

For the purpose of practical illustration, the arrangement represented by Fig. 45 is inserted above, being a patented disposition of detail by Messrs. Day and Co., of Southampton, in 1863, especially referring to the employment of the ordinary suction air pumps of marine engines, for the purpose of surface condensation, in such manner that one pump draws from the condenser the condensed or fresh water for the supply of the boilers, and that the other pump draws from the condenser the sea or condensing water which it discharges overboard. This illustration shows the plan half in section half complete. The elevations are shown also complete and in section: the internal portion relating to the pumps only. The plan represents the condensers beyond each steam cylinder, near the sides of the hull, the pumps being centrally situated as for the injection system. The steam, it will be seen, enters the cylinders through the inner trun-

Fig. 45.



MESSRS. DAY AND CO'S OSCILLATING ENGINES AND SURFACE CONDENSERS.

nions, and discharges through those on the extreme sides of the arrangement, and thence to the condensers. The pipes and stop valves are suitably arranged, and the position is in accordance with the requisitions. The complete elevation shows one condenser—the tubes being dotted—the discharge and supply pipes, air vessels and pumps, also the entablature and steam cylinder. The sectional elevation represents each pump in section, also the discharge chambers, and the vertical stop valve boxes complete,—seen also in the plan at the sides of the hull.

The motion for the pumps is derived from the intermediate cranked shaft, or eccentrics can be adopted if preferred. The action of the pumps is as follows: the air pump is on the same side of the centre line as the condensers, or between the same. The steam enters the condenser at the top of the tube compartment —the connection is seen in the elevation—after passing through the tubes the condensed steam passes through the passage—seen in section in plan—to the air pump, the arrangement of the valves being depicted in the sectional elevation. The discharge is effected into the chamber over the pump, and thence into the sea, if requisite, by the pipes connected to the ship's side: but generally into the boilers. If separate or feed pumps are adopted, they are connected to the opening shown in the bottom of the discharge chamber. The circulating pump is situated opposite to that last alluded to, the connection with the discharge chambers and disposition of the valves being also similar. The sea water flows into the upper portion of the condenser, through the pipe shown in the plan and elevation; the drainage being through the pipes, also seen in the plan and elevation. The arrows shown in the elevation, indicate the line of the water's circuit through the condenser. The final discharge is effected through the discharge pipes, over the pump, connected to the ship's side.

In the event of fracture or leakage of the condensing tubes, or the conversion of the surface into an injection condenser, it is thus occasioned. The stop valves for regulating the circulating water connected to the pipes leading from the condenser are closed—as shown in section in plan. Injection water is admitted through the cock or valve—shown in plan and elevation—connected to the condenser above the tubes. This water meeting the steam from the trunnions, condensation ensues, and the air pump drains and discharges as before. A communication with the circulating pump is effected by the removal of the internal door—shown in the sectional elevation—at the side of the air pump near the foot valve. By this reverse adoption of the several passages, both pumps perform the same duty, and the system of condensation altered accordingly. The utility of the remaining stop valves is so obvious, that a notice of them is unnecessary.

This arrangement is novel and efficacious, but at the same time wanting perfection of situation for the pumps, in relation to the condenser. It will be remembered that the former are at the centre of the hull, but the latter at the sides of the same. Now with engines of moderate power this arrangement may not be deemed objectionable, but in large engines the distance between the suction valves and the

condenser will always lower their action, to say nothing of the "back-wash" always under the condensing tubes. A partial remedy for this can be caused by putting a second set of valves vertically, secured near the condensers in the passages leading from the same to the pumps. This, although not shown in the drawings, and not alluded to in the specification, has doubtless also occurred to the patentees. The remaining portion of the arrangement is of the ordinary kind, as the patented disposition does not interfere with the details, except where especially alluded to.

The firm in question have fitted the mail steamer "Syria," belonging to the P. & O. Company, with engines and condensers as those illustrated, in the year 1866.

The collective nominal horse power is 450, and diameter of each cylinder 76 inches, with a length of stroke of 7 feet. The surface condensers have been very successful, producing a vacuum of 27·5 inches, with a temperature of 130° Fahr. for the feed water.

Messrs. J and G. Rennie about the same time fitted the "Nyanza," a sister mail steamer to the "Syria," belonging to the same Company, with surface condensers and oscillating paddle engines. The condensers are arranged in the side wings of the engine room, and the air pumps centrally of the engines, as for the ordinary injection condensers—which latter are fitted also in case of requirement. In the place of a crank on the intermediate shaft, the firm in question preferred eccentrics of a throw of one foot $4\frac{1}{2}$ inches to impart motion to the air pumps, each pump being 3 feet $10\frac{1}{2}$ inches in diameter. To accomplish this, the eccentrics are each 5 feet 2 inches diameter, the intermediate shaft being at that part 22 inches diameter. The eccentrics are doubtless the largest yet constructed. The surface condensers are tubular, both containing 8,700 tubes; each tube is $\frac{9}{16}$ of an inch in diameter; the total lineal length being 43,000 feet. The engines are 450 horse power nominal collectively, the cooling surface of the condensers are therefore 14·06 square feet per nominal horse power. The diameter of each steam cylinder is 6 feet $6\frac{7}{16}$ inches, and the length of stroke for the piston 7 feet. The diameter of the steel piston rods are $9\frac{3}{4}$ in., and the crank shaft — wrought iron — at bearings 18 inches. The paddle wheels are 27 feet diameter, with feathering floats each 10 feet long and 4 feet 6 inches wide.

With a mean pressure of steam at 25·625 lbs. on the square inch, vacuum 26·125 lbs., the engines attained 24·875 revolutions per minute, producing 2,600 horse power collectively, and propelled the vessel 13·5 knots per hour.

The circulating water is forced through the condenser by centrifugal pumps and separate high speed engines.

Messrs. James Watt and Co. lately fitted oscillating engines and surface condensers in a new Brazilian steamer, 260 nominal horse power collectively. The position of the condensers is centrally of the hull transversely, and directly at the back of the lower frame longitudinally. This arrangement, it will be noticed, is unlike that with previous arrangements alluded to. Surface condensers are fitted with horizontal double

acting air and circulating pumps under them. The motion for the pumps is derived from arms secured to the trunnions or gudgeons of the steam cylinders.

Messrs. Ravenhill and Hodgson have lately repaired the S.S. "Ripon," of good repute, and introduced surface condensers in the wings of the engine room. The air pumps are each worked by eccentrics, and the water is forced through the condensers by separate centrifugal pumps; the motion for these pumps being derived from the crank shaft by mitred gearing on it, and at each extremity of the counter shafts and pump spindles.

There are other means proposed to obtain a vertical action for the pistons of the pumps in question, and dispensing at the same time with the cranks or eccentrics on the intermediate shaft, viz., by projecting arms on the upper portions of the cylinders. This idea has been patented by Messrs. Davison and Paterson, and described in their specification thus, in 1861 :—

"In the case of oscillating engines, the air pumps and circulating pumps are worked by the motion of the cylinder. The pumps are worked by means of arms or levers attached to the cylinder; and to sustain the strain arising from thus working the pump, guides are attached to the cylinder cover, between which the piston rod slide block works, or the piston rod may be otherwise suitably guided. In some cases we arrange a supplementary pump, in connection with the valves and passages of the circulating pump, with provisions for working it by auxiliary power, for the purpose of continuing the action of the condenser when the main engines are not working. Such supplementary pump may be placed in the chamber of the main pump, or it may be placed apart, and communicate therewith by pipes or passages; in all cases, however, the valves of the main pump act also for the supplementary pump. Or, instead of such arrangement, provision may be made for disconnecting the main pump from the main shaft, and for driving it by auxiliary power."

The surface condensers in this case are placed directly beyond on each side of the cylinder, fore and aft of the same, the pumps being single-acting of the ordinary kind. It will be noticed that the inventors especially recognize the effect of the pump's connection with the cylinder by guiding the piston rod of the latter. Doubtless a better mode will be to dispense with the idea altogether, and work the pumps as in the example alluded to in page 201, and represented by Fig. 45. While in some instances a horizontal arrangement is preferred, and likewise a double-action.

Now with double-acting pumps the diameters are lessened, but the valves increased in number, and thus the horizontal action often deemed preferable to a vertical. A compact horizontal arrangement for the pumps can be attained by working them from the lower ends of the steam cylinders, centrally. With a stroke of four feet for the steam piston, a motion of two feet for the pumps can be effected, by putting the centre of the swing bar four feet from that of the trunnion; the bar being secured to the centre of the cylinder. A shorter stroke can be given to the pumps, while raising them also, by reducing the length of the bar.

The circulating and air pumps can both be worked by a cross-head, with guide blocks and channels for each rod, to preserve a direct motion. By this mode of guidance, trunks will be dispensed with, and a single connecting rod only requisite to each cylinder. The condenser can be sustained above the pumps in one compartment; all the suction valves also can be inverted above the pumps, and those for the discharge similarly placed. Now it may be argued that this mode of connection will be productive of undue strains on the steam piston rod; inasmuch, that when the steam piston is at full stroke, those for the pumps are at half their travel, the power of the former being therefore the least, and the requisition for the latter the most. Further, it may be said, that the motion of the steam piston is the greatest when it is at half stroke, and the labour of the pumps just then commencing; also, the power gradually lessening when actually it should be increasing. Notwithstanding these queries, which are worthy of due attention, by correctly balancing the cylinder on the bearings, to meet the requirements of the case, the evil alluded to can be greatly mitigated.

SLIDE VALVE LINK MOTION, AND STARTING GEAR.

The arrangement of the detail requisite to impart the motion to the slide valve forms an important consideration with the marine engine, and more particularly when the oscillating type is dealt with. The cylinder vibrates, and also its appendages, and the main accomplishment is a vertical motion for the slide valves, independently of their lateral motion. To produce this a sliding quadrant is requisite, and the efficiency of this detail depends on the curve of the slot. The correct shape of this curve is undoubtedly semi-elliptical, the lesser radius being at the extremities. For practical purposes, an uniform curve is usual, the radius of which is—the angular distance between the centres of the sliding blocks and the trunnion, when the working levers are on the horizontal line. The illustration, Fig. 46—page 206—is an arrangement of link motion and starting gear of the oscillating engines constructed by Messrs. James Watt and Co.

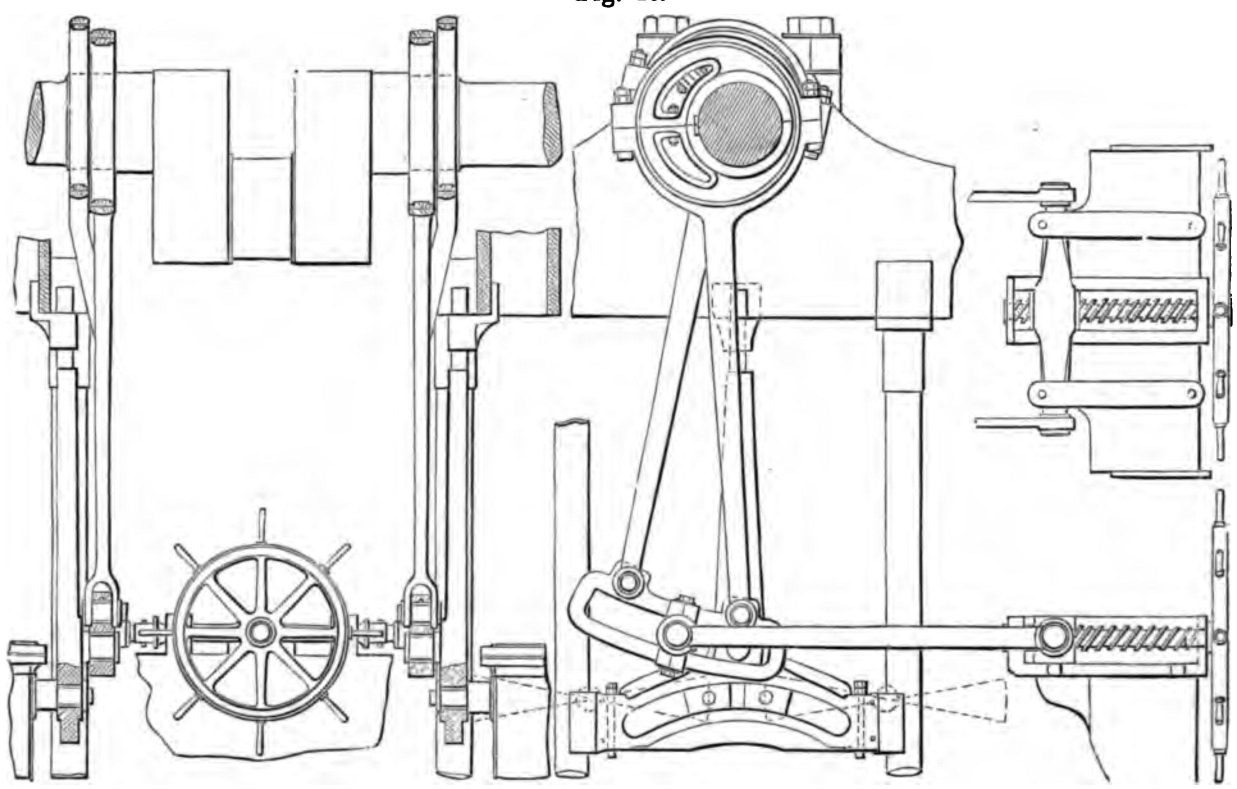
The eccentrics are in halves, connected by bolts; the rods being of the ordinary kind, with double eyes to clasp the link. The peripheries of the eccentrics are fitted with brass hoops with projections to suit the recesses formed in the bands of the rods, the hoops being also recessed on the eccentrics to prevent lateral displacement. The link is the usual shape, with brasses in the eyes, adjustment being caused by cotters above the top brasses. The sliding quadrant is guided at the sides by semicircular brasses clasping the main columns, and adjusted by keys and set nuts; the guidance of the upper end is attained by a bracket, secured to the entablature. The side elevation shows the link thrown back, or at full stroke for starting, the quadrant and the levers being at half-stroke, the latter being indicated by dotted lines only. The end elevation shows the links in section, also a portion of each of the quadrants, the eccentrics being represented as keyed on the intermediate crank shaft.

Next, attention must be given to the "start-

ing gear." The requisition of this gear is to raise or lower the slide valves without the motion of the cranks, or steam pistons. Now, with the motion as arranged, this is attained by moving the link forward or backward as may be required, and the fixed points of the eccentrics cause the valves to move simultaneously. Many arrangements are made and carried out to meet the requirements now alluded to. In some cases the shifting rod is connected to a lever, the latter being keyed on a weigh shaft, and motion imparted by a worm and toothed quadrant. In other instances, a pinion and wheel have been preferred; while a third example dispenses with the wheel, and the rod is connected to a shifting rack, the latter being worked by the pinion. Each device, it is noticed, accomplishes the same final result, although the mechanical contrivances are different.

In the example under especial notice and illustration, the arrangement is unlike any other alluded to. The principle of the motion is a direct action, this being attained by a revolving screw and sliding cross bar. The side elevation shows the screw supported in a frame at each end, with the starting wheel beyond the outer bearing. The cross bar is also shown, but more clearly in the plan of the gear. The connection with the links is attained by flat rods hung at each end of the bar. In order to counteract the spiral action of the screw, and preserve a direct motion for the bar, two side guides are

Fig. 46.



MESSRS. WATT'S LINK MOTION AND STARTING GEAR FOR OSCILLATING ENGINES.

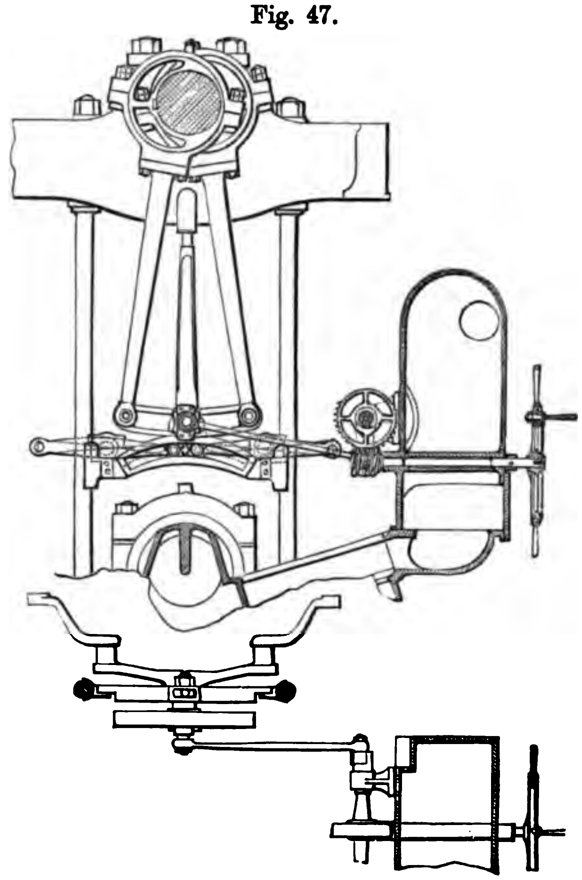
used, and a central slipper guide directly under the screw. It will thus be understood that, on imparting a rotatory motion to the screw by the hand wheels, the cross bar receives the effect, and a sliding motion is imparted, by which means the link is shifted in the desired direction. The end elevation represents the position of the wheel between the cylinders, and its height from the trunnion centres. This arrangement is used by many firms besides that under allusion; Messrs. Rennie, Maudslay, and engineers in Scotland, having adopted it. The example illustrated being of notice as a guide for future practice, the following dimensions are compiled from the working drawings, from which the engraving was reduced :—

	Ft.	In.
Motion for Slide Valves	0	7
Diameter of Working Lever's Stud	0	4
Diameter of End Pins	0	2½
Direct length of Levers from Stud	1	7
Radius of Slot in Sliding Quadrant	2	6½
Width of Slot	0	4
Thickness of ditto	0	4½
Radius of Link	7	6
Width of Slot in Link	0	4½
Thickness of Link	0	4
Diameter of Block Pin	0	3
Travel of Sliding Block	1	9½
Length of ditto	0	8
Diameter of Eccentric Rod's Pins	0	2½
Diameter of Band Bolts	0	1¾
Diameter of Starting Wheel Shaft	0	3½
No. of Threads in one pitch of Screw, 4.		
Width of Side Guides	0	4
Width of Slipper Guide	0	8¼
Width of Connecting Rods	0	3½
Thickness of ditto	0	1½
Diameter of Starting Wheel to centre of the handles	3	6

The means for shifting the link, as already noticed, can be accomplished by various dispositions of the details. The arrangement just alluded to is a "direct action," or a sliding motion for the propelling end of the connecting rods. Now, while admitting that this is the correct motion, it must not be forgotten that complication of the working parts is introduced, also expense of manufacture and repair the result. To mitigate these evils is the aim of all right thinking engineers, and the firm in connection with the example alluded to is not behind in this matter.

In order to reduce the number of the details, and lessen the expense of construction, the

Fig. 47.



MESSRS. DUDGEON'S LINK MOTION AND STARTING GEAR FOR OSCILLATING ENGINES.

arrangement depicted by Fig. 47 is often preferred. This is an example of the late general practice by many eminent firms, one of them

being the Messrs. Dudgeon, from whose working drawings those engraved were compiled and produced. This arrangement is termed the worm, wheel, and lever motion, which the following description will pourtray. The side elevation shows the eccentric link, sliding quadrant, and working levers at half stroke; the entablature and trunnion block being shown complete, beyond the gear. A portion only of the condenser is represented in section; but a complete sectional elevation of the discharge chamber is depicted, to show the means for supporting the starting wheel shaft, which is a tube cast with the chamber, and a loose collar on the outside prevents a lateral movement. The hand wheel is keyed on the shaft, and the horizontal projecting handle on the wheel is for the purpose of quick manipulation, when required. The shaft has formed with it, at the other extremity, a coarsely pitched screw, generally termed a worm. A toothed quadrant gearing with the worm motion, at right angles to that of the starting wheel, is acquired. The quadrant is keyed on a single shaft, the latter being supported in brackets, secured to the side of the chambers. At the ends of the shaft, levers are keyed and connected to the links by rods, thus any motion given to the starting wheel is transmitted to the links.

The location of the working levers and quadrant, in relation to the starting wheel, is seen in the plan, one set only being represented. The levers are supported by plummer blocks—seen in elevation only—secured to provisions formed on the cylinders. The dotted curves seen in the elevation of the chamber are explained in the plan, being the spaces requisite for the action of the motion levers. The principle of this arrangement is so obvious from the drawings, that it is almost needless to state that, on motion being imparted to the wheel, the worm acts on the toothed quadrant, and the levers, by their connection to the link, shift the same, and thus the desired effect is attained.

The following dimensions are the practice of Messrs. Dudgeon, for a pair of engines 200 horse power nominal collectively:—

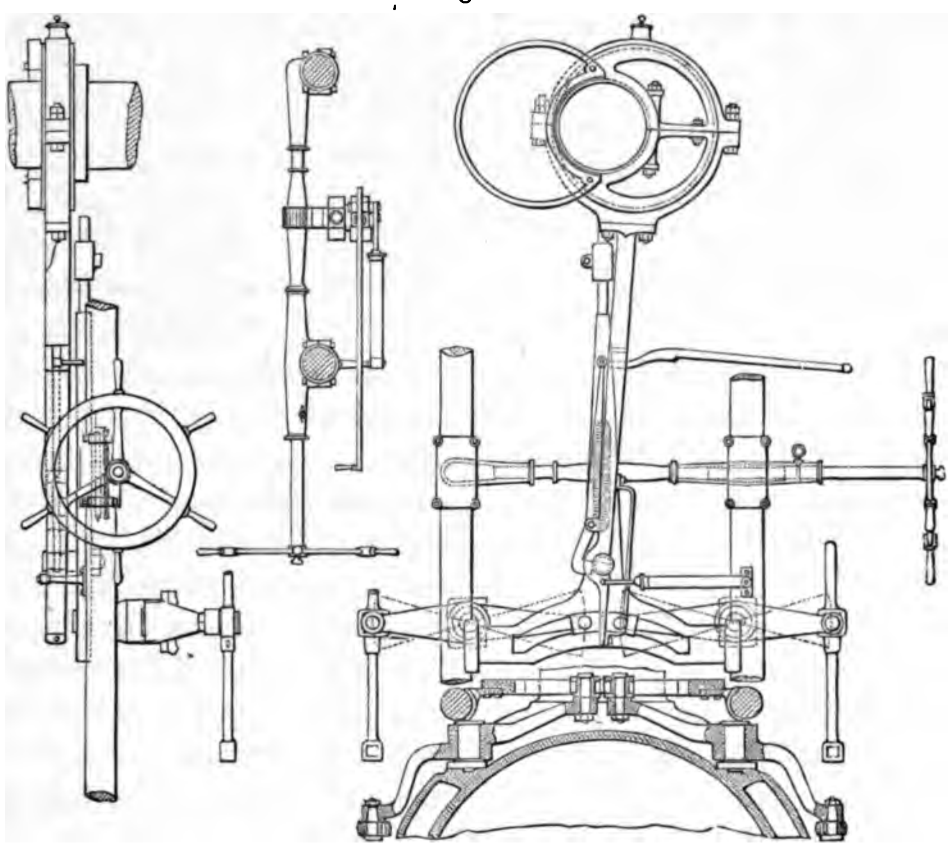
	Ft.	In.
Motion for Slide Valves	0	5¾
Diameter of Central Bearing of Working Lever	0	4
Diameter of End Pins	0	2
Direct Length of Levers from Bearing	1	3½
Radius of Slot in Sliding Quadrant	2	3¾
Width of Slot in ditto	0	3¼
Thickness of Slot in ditto	0	3
Radius of Link	5	6
Width of Slot in Link	0	3½
Thickness of Link	0	2½
Diameter of Block Pin	0	2⅝
Travel of Sliding Block	1	6
Length of ditto	0	5½
Diameter of Eccentric Rod's Pins	0	2
Diameter of Band Bolts	0	1¼
Diameter of Starting Wheel Shaft	0	2
Radius of Toothed Quadrant	0	7⅜
Diameter of Weigh Shaft	0	3½
Diameter of Worm	0	4
Diameter of Connecting Rods	0	1¾
Diameter of Starting Wheel to centre of handles	3	0

SINGLE ECCENTRIC MOTION AND STARTING GEAR.

To correctly understand any branch of science or arrangement of mechanism is first to become acquainted with the requirements. From this cause the engineer develops all his productions, and obviously also whether one, two, or more attainments are required at once,

each matter is alike in principle. The motion for the slide valve is a simple effect to be produced, but the main question is the best means for preventing the action when such is desired. The link, when at half travel, brings the working levers on the horizontal line, and thus the slide valves cover the ports, but the speed of moving the slides entirely depends on the power expended to shift the links. Another fact also presents itself for notice; when the eccentrics are in a certain position the link cannot shift the valve to produce much effect, and thus one engine actually has to start its neighbour. Now to obviate this, the entire disconnection of the slide valves from the eccentric motion is required, and thus the arrangement represented below, by Fig. 48, is often used by some authorities in preference to those examples having had prior notice. To clearly understand this arrangement, the main effect must be remembered, viz., independent means of working the slide valves. In this case the slides can be shifted to the full, or any portion of the stroke, irrespective of the angle of the eccentric or position of the steam piston, and thus the engine can be held—so to speak—without any regard to the mechanical means for working the slides. Now, to render obvious this matter for practical purposes, the following description is given in connection with the drawing alluded to.

Fig. 48.



SINGLE ECCENTRIC MOTION AND STARTING GEAR FOR OSCILLATING ENGINES.

The side elevation represents the eccentric at half stroke, also the sliding quadrant and working lever. The eccentric has attached to it a portion termed a counter-balance, the utility of which is that the eccentric—being loose on the shaft—retains a horizontal or equilibrium position when the rod is disconnected from the sliding quadrant. For the purpose of fixing the eccentric at the requisite angle, for going ahead or astern, stop pins are inserted in the crank shaft, projecting beyond the periphery. A provision on the boss of the eccentric bears against either pin when the rod is in gear; the direction of the rotative movement causing the constant contact of the pin and eccentric boss. On disconnecting the eccentric rod, the eccentric retains the equilibrium position—already alluded to—and remains stationary until the pin comes in contact with the provision in question, by the rotation of the shaft.

Next, attention must be given to the starting gear. Presuming the engines are to be stopped, the hand lever—level with the top of the starting wheel—is pushed towards the eccentric rod, and the lower end of the lever—being cranked into a slot in the rod—forces the rod from its bearing on the quadrant pin. The pin then works in the loop secured to the front side of the rod, and the cranked end of the lever prevents the eccentric rod returning to its bearing. When the engines require to be started, the hand wheel is used as a preliminary means for working the slide valves. Now, assume that the eccentric rod is disengaged as described, the action of the slide valves is independent of the eccentric. The position of the starting wheel will be as that seen in the drawing under notice; although the eccentric rod is shown connected,—this is only for the means of representing the application of the gear in question. On referring to the end elevation, it will be noticed that on the starting wheel shaft a pinion is secured, and this gears into a rack, the latter being secured to the sliding quadrant. Now, on turning the pinion, the rack ascends or descends, as may be required, and thus a movement is imparted to the slide valves similar to that derived from the eccentric. The plan of the starting gear, between the elevations, conveys a ready conception as to the connection of the several portions of the detail, and the sectional plan of the working gear—under the side elevation—depicts the angles requisite for the levers and position of the quadrant, and means for guiding the same.

Having thus far become acquainted with the requisitions and the mechanical means to acquire the same, attention devoted to the advantage of the arrangement will not be misapplied. Suppose a ship, fitted with oscillating paddle engines, and single eccentric motion and starting gear, as shown, is required to be reversed in her way as speedily as possible, the application of the gear in question will be thus apparent. The eccentric rod is thrown out of gear—as explained—and the starting pinion pushed in gear with the rack, the quadrant imparts the required motion to the slide valves, to cause the steam to enter the cylinders at the opposite ends—the direction of turning the wheel of course determining this latter effect. On the engine starting, the eccentric rod can be set on the

quadrant pin, as soon as the latter is level with the opening, and the spring—contained in the box connected to the entablature column—retains the rod in its bearing. The pinion, on the starting wheel shaft, being withdrawn from the rack on the quadrant, the eccentric becomes the agent for working the valves, without affecting the wheel in question.

When the engines are required to be brought up suddenly, as in the event of a collision or any other cause, the disconnection already alluded to ensues; and, on the starting pinion gearing with the rack, the slide valves can be instantly set at half-stroke, thus preventing any admission of steam into the cylinders.

The main advantage in question is, therefore, that the engines can be handled by separate gear, at any point of the stroke of the piston, and both pistons can be operated on by the steam equally for the purposes required. When the slides are worked by the motion of the eccentric, and disconnection is not permitted, the speed of the pistons determines that for the valves; but when hand, or separate power, shifts the slides, the admission or stoppage of the steam is more readily acquired. Large engines, fitted with the gear under notice, can be started, stopped, and reversed almost instantaneously, without straining any portion unequally.

It is preferred, in some instances, to dispense with the spring and box—for the purpose of holding the eccentric rod on the quadrant pin—and adopt a lever with a catch spring, connected to a slotted quadrant: the latter being secured to the side of the discharge chamber of the air pump—the starting wheel, in this case, being beyond the chamber. With this arrangement, the horizontal hand lever with the cranked end is dispensed with, also the slot in the eccentric rod, and the connecting rod is attached to the back of the loop instead of the side of the rod.

For engines of small or moderate power, or between 10 to 50 horse power nominal collectively, the hand wheel is not requisite, as the slides can be shifted by long double levers; the ends of the lesser lengths being connected to the quadrants. During the motion of the slides the starting levers are likewise affected, which, for small engines, is not of much importance. The adoption of either the wheel or lever for manipulation depends on the power requisite to shift the slide valves. The less the friction, therefore—due to the action of the steam—the less the power requisite to start, stop, or reverse the engines.

The firms at present most partial to the arrangement illustrated are Messrs. Penn, Messrs Maudslay, and Messrs. Ravenhill. Each of those authorities differ slightly in their notions of design, but in principle of motion all of course accomplish the same purpose. The example illustrated is very similar to the practice of the latter firm.

To make the engraving of practical utility, the following proportions are compiled from the single eccentric motion and starting gear lately fitted to a pair of engines, of 250 horse power nominal collectively:—

	Ft.	In.
Stroke of Slide Valve	0	8½
Diameter of Slide Valve Rod . . .	0	1⅝

	Ft.	In.
Throw of Eccentric	0	10
Diameter of Eccentric	2	1½
Diameter of Crank Shaft	1	1½
Diameter of Central Connecting Bolt of Eccentric	0	1⅜
Ditto Outer ditto	0	0¾
Ditto Bolts securing Counter-balance	0	1
Ditto Hand Bolts	0	1¼
Width of Band	0	2¾
Thickness of Band	0	1⅜
Radius of Counter-balance	1	1¼
Thickness of ditto	0	1⅝
Diameter of Eccentric Rod	0	2¾
Length of Hand Lever	3	3
Length of Loop for Quadrant Pin	0	11½
Diameter of Spring in Box	0	1
Vertical distance between Centres of Crank Shaft and Quadrant Pin—at half-stroke of Valve	5	10¼
Radius of Slot in Quadrant	2	4⅛
Width of Slot	0	2⅞
Length of Chord of Slot	2	0⅜
Thickness of Quadrant	0	2½
Diameter of Quadrant Pin	0	2¾
Length of Quadrant Guides	0	9
Diameter of Working, Lever Main Stud	0	3⅛
Ditto ditto Pins	0	1¾
Length of Swing Bearing of Lever	0	5¾
Diameter of Boss ditto	0	6¾
Outer Length of Lever	1	6⅞
Inner ditto	1	3⅝
Diameter of Starting Wheel	2	0
Length of Handles	0	6
Diameter of Shaft	0	1¾
Distance between Centre of Pinion and Starting Wheel	4	2
Diameter of Pinion	0	3¼
Pitch of Teeth	0	1
Number of Teeth in Rack, 14		
Breadth of Teeth	0	2½
Diameter of Securing Bolts	0	0¾

EXPANSION VALVES AND GEAR.

Steam, being very elastic, naturally admits of great compression—hence the gain when using it expansively. It is universally known that the greater the pressure the power will be increased in proportion. Now let it be presumed that steam is required for a cylinder at the rate of 100 cubic feet per stroke of the piston, at a given pressure throughout, thus imparting to the piston a certain power, it is obvious that a given quantity of steam must be evaporated consecutively from the boiler; leakages, friction, reduced temperature, &c., being also considered.

Let it be assumed, also, that a given quantity of fuel is consumed in proportion to the power attained; next, suppose it to be deemed necessary to reduce the consumption of the fuel in the next example, or, rather, pair of engines. The first consideration of the improver, or rather the designer, would be that the power must not be reduced. This then produces the idea, that to increase the pressure of the steam, imperatively increases its elastic force. Thus it is clear that by increasing the pressure, a reduction in cubic contents of steam from the boiler is attained, requisite for each stroke of the piston; and, consequently, the evaporation required is lessened in proportion. Now, it must be strictly understood that, to produce correct expansion, cubical capacity in the cylinder must be observed. Thus, for example, a cylinder of a given capacity, with a certain grade of expansion, must be of relative cubic contents, in proportion to the amount and pressure of the steam admitted. While if the steam is increased in pressure, and the same grade of expansion retained, the cubical contents or area must be enlarged, to exhaust at the same pressure and temperature as before.

To use steam economically, is to reduce the temperature of it as low as possible before it enters the condenser, and thus affect the time of condensation, producing thereby a better

vacuum than if the temperature was higher. It should be remembered, also, that each volume of steam from the cylinder must be perfectly condensed before the succeeding volume enters.

Having thus briefly alluded to the principles of economy, let it be next considered what is the assumed action of the steam while in the cylinder, *i.e.*, from the admission to the exhaustion. Presume the piston to be at full stroke, or at the top or the bottom of the cylinder, the slide valve should have a lead, or be open for a given portion. Now it must be noticed that the steam was admitted before the piston had completed its stroke, consequently compression and partial, if not total, stoppage of the admission must have ensued. Trivial as this may appear, yet in high velocities and full leads the indicator diagram shows a round corner, commonly known as cushioning. The lead, however, must not be discarded, as it acts as a spring between the piston and the ends of the cylinder, a matter of great essentiality in short strokes, and, as before stated, high velocities.

To return to the piston at full stroke, presume the crank to have passed the dead centre; the steam will then act with force due to its admittance, so that it is not erroneous to assume that the greatest effect of the steam on the piston is when the valve is at full stroke, or the port widest open for the supply. The piston, be it remembered, is now supposed to be exerting its utmost efforts; due, of course, to the action of the steam. Presume the valve to return and close the port at a given point of the stroke of the piston. At this point expansion commences, the piston being propelled after that by the elasticity of the steam; hence the gain in economy before alluded to. The time the steam is admitted into the cylinder is while the valve is moving forth and back, or opening and closing the steam opening. The time the steam commences its actual propulsive power on the piston to that of its cut off = width of opening $\times$ 2 − lead. The valve, when last alluded to, was presumed to be at the edge of the port, expansion therefore being in full operation. Before the steam can exhaust, the valve must open the same port, but contrary in its action. The time ensuing between the point of cutting off the supply and opening the port for the exhaust is, of course, due to the outside and inside laps. Therefore the time allowed for expansion = outside lap + inside lap. This will be better understood by presuming the valve to have no laps: it will then be obvious that if additional length be added to the valve, that increase must be observed. The time allowed for exhaustion is due to the speed or travel of the valve from a point to a point, or in principle as before stated for supply and expansion. Now, presuming the valve has covered the port, and travelled so that the inner part is at the outside edge of the bar, it can readily be understood that exhaustion must ensue instanter with the action of opening the port. The valve now moves forward and backward for a given length, occupying thereby a certain time. If there was no lap inside, it is obvious that when the valve was at half stroke, the time for exhaustion would be while the valve was opening or closing the port or opening, on the opposite end of the cylinder, outside lap being also

considered. When there is an inside lap, the valve must move from the half stroke that sum or distance before the exhaustion commences; the valve then moves back, due to the distance as before stated, and forward the same, less the inside lap. The formula for time of exhaustion, therefore, will be thus, outside lap + width of opening caused by valve × 2.—[inside lap × 2.]

By these formulæ—time being considered—an almost perfect knowledge of the action of the steam can be attained, its elastic force and the volume consumed at each stroke of the piston known. There may be, perhaps, existing doubts as to the application of the rules now noticed, also their practical utility; a little further elucidation will, therefore, not be out of place. For example, presume a pair of engines of 150 horse power nominal collectively; length of stroke, 4ft.; diameter of cylinder, 48⅜in.; utmost grade of expansion, $\frac{1}{6}$ of the stroke. Let it be assumed the travel of the valve is 10 in. unalterable. The cubic contents of the cylinder will be deduced in the usual form, $1837 \cdot 93 \times 48 = 88220 \cdot 64$ cubic contents in inches. The space occupied by the steam at $\frac{1}{6}\text{th} = \frac{88220 \cdot 64}{6} = 14703 \cdot 44$, or $1837 \cdot 93, \times 8 = 14703 \cdot 44$. This, then, shows that the expansive powers of the steam must emanate from a volume of steam whose capacity equals $\frac{1}{6}$th of the total contents of the cylinder. Now the amount of expansion is, of course, due to the capacity occupied after the termination of the admission of the steam: to define, therefore, the amount of expansion, time and space must be duly considered. The piston in the present example is presumed to have moved 8 in., or $\frac{1}{6}$th of the stroke. The valve has closed the port, and has to travel the outside lap + inside lap, until the steam is released from the cylinder. To further demonstrate the present theory, again time must be noticed. Let it be assumed that the engine, or rather the piston, moves at the rate of 50 strokes per minute—two strokes, of course, making a revolution. The valve, be it remembered, moves at the same speed, action only being now alluded to. Now, when the piston is at the extremity of its stroke, the valve has performed the greatest portion of its movement to the utmost point, and, consequently, in a given time moves in a contrary direction. The relative positions of the valve to the piston are due to the arcs passed through rotatively, each being the same in proportion.

Thus far it will be noticed the action of the steam is presumed to be governed by the slide valve only; and to cause different grades of expansion, obviously valves of different laps are requisite. Now, as this would be expensive in material as well as tedious in operation, a separate valve is generally used, located as near the slide valve as practicable. Another cause for the use of the expansion valve, is that the steam can be admitted slowly and cut off suddenly; whereas, with the slide, these operations are nearly alike, as far as speed of action is concerned.

Although the actual gain by using steam expansively has been explained to a certain extent at the commencement of this section, it will not be out of place—before describing the mechanical arrangements in connection—to further treat of the matter in question.

Assume, for example, a cylinder with a six-

feet stroke of piston. Now, it is required to use six grades of expansion—which, as already stated, with the slide valve, six different outside laps are requisite. When the steam is admitted into the cylinder, imagine the pressure to be 30lbs. on the square inch. On the piston moving one foot the cut off occurs, and expansion commences. The termination of the expansion is due to the position of the slide when the steam was interrupted. For the purpose of further exemplification, imagine the piston to have moved two feet, when the exhaustion of the steam ensued, the pressure will be about 14lbs. on the square inch—but theoretically 15lbs., the loss of pressure being due to the reduction of the temperature. Again, assume the cut-off to occur when the piston has proceeded two feet, twice the amount of steam will be admitted to that in the previous case. The piston is being propelled by the expansion as before, until the release ensues, which presume to be four feet, from the cubical space being doubled, as before, the pressure will be reduced proportionately the same also. Now if the steam is cut off at four feet stroke of piston, and released at five feet, $\frac{1}{5}$th of the pressure only is lost, or the steam will be about 24lbs. on the square inch when exhausting. The most economical mode of using the steam will be to cut off at $\frac{1}{8}$th, and exhaust at the same distance from the opposite end of the stroke. The steam will then be expanded four times the original capacity, and reduced proportionately in pressure when exhausting at about 7lbs. on the square inch.

Mr. Fairbairn, when treating on this subject in his "Useful Information for Engineers," imparts a little valuable information on this important subject, which it has been thought worthy to introduce. He states: "If we take a cylinder of any given diameter, say five feet long, and divide it into five equal parts, we shall then have the different equidistant positions of the piston, as it ascends from the bottom to the top of the cylinder. Suppose the first space to be filled with steam at 40lbs. on the square inch, the piston to be moved one foot. Now, it is obvious that if we cut off or interrupt the flow of the steam at this point, that the next foot of the stroke will double the space occupied by the steam; and there being no further supply from the boiler, the steam will have to expand itself into double its original volume, and its pressure by this dilatation will be reduced from 40 to 20lbs. on the square inch. The piston having moved at this point with the force then reduced to 20lbs. on the square inch, it again moves forward another foot, when the original space occupied by the steam becomes enlarged three times, with a proportionate decrease of the pressure in the steam; that is, with a pressure of one-third of 40lbs., or $13\frac{1}{3}$lbs. acting upon the piston, and so on to the other points of the stroke. The pressure of the steam at the successive equidistant intervals of the stroke will be as follows: 40lbs., 20lbs., $13\frac{1}{3}$lbs., 10lbs., and 8lbs. These pressures, derived from the law of Marriotte, are no doubt slightly in excess, inasmuch as the vapour suffers a loss of temperature upon expansion.

"The deductions to be made for this loss of heat and loss of pressure in the process of working can be ascertained by the *indicator*; but for our present purpose it will be sufficient to assume that there is no loss.

"In order to find the work performed by the steam in one stroke of the piston, we shall first find the mean pressures of the steam acting through the successive intervals of the stroke; and then from these mean pressures we shall find the total mean pressure.

Pressure in the 1st foot of the stroke			.	= 40 lbs.
Mean ditto 2nd foot of the stroke			$= \frac{1}{2}(40 + 20) =$	30 "
" 3rd	"	"	$= \frac{1}{2}(20 + 13\frac{1}{3}) =$	$16\frac{2}{3}$ "
" 4th	"	"	$= \frac{1}{2}(13\frac{1}{3} + 10) =$	$11\frac{2}{3}$ "
" 5th	"	"	$= \frac{1}{2}(10 + 8) =$	9 "
				5 | $107\frac{1}{3}$ lbs.
Total mean pressure		.	.	$21\frac{7}{15}$ lbs.

$\therefore$ The work in one stroke = the mean pressure $\times$ the length of the stroke $= 21\frac{7}{15} \times 5 = 107\frac{1}{3}$.

"Now, when the steam acts uniformly throughout the whole of the stroke, the work $= 40 \times 5 = 200$. But this work is done with five times the quantity of steam that is employed when acting expansively, therefore the work done by an equal quantity of steam is the fifth of 200, or 40. Comparing the numbers $107\frac{1}{3}$ and 40, we find that the steam used expansively performs $2\frac{1}{2}$ times the work that it does when it is used non-expansively, or with a constant pressure.

"This simple method of calculating the work performed by the expansion of the steam gives the result a little in excess. The following method, depending upon Thomas Simpson's rule for finding the area of irregular curved surfaces, is more exact.*

Work done expansively $= \frac{1}{3}[40 + 9 + 4\ (20 + 10) + 2 \times 13\frac{1}{3}] = 65\frac{2}{3}$.

Work done before the steam is cut off $= 40 \times 1 = 40$.

$\therefore$ Total work in one stroke $= 65\frac{2}{3} + 40 = 105\frac{2}{3}$, which corresponds very nearly with the work as before found.

"In this calculation some allowance must be made for the loss of heat, consequently the loss of pressure, during the process of expansion, which may be ascertained by diagrams taken from the indicator. This loss of heat by expansion is much greater than is generally imagined, as we seldom find general practice to agree with deductions derived from theoretical calculations."

Having thus far treated of the laws of expansion, attention must now be directed to the valve and gear generally adopted. It is obvious that the main consideration is the time allowed for the expansion, and to attain this an unequal movement for the valve is requisite. To produce this effect, the well-known cam motion has had universal favour, and doubtless with engines moving at a moderate number of revolutions, or from 30 to 40 per minute, it cannot be excelled.

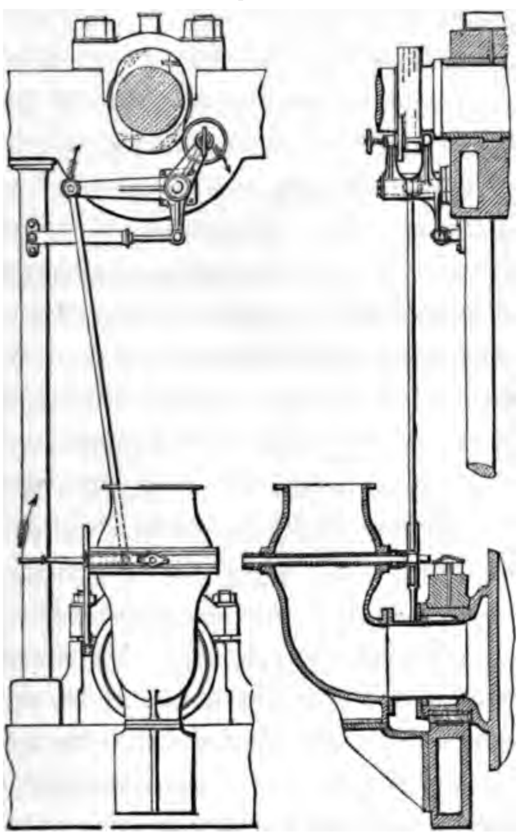
The illustration, Fig. 49—page 217—is an arrangement of expansion gear and throttle valve, as at present adopted by many eminent firms, both in London and Glasgow.

There are of course, in each maker's example, minor deviations and additions, but the general arrangement and principle of action is pretty much the same as that illustrated. For example, in some instances it is preferred to attach the end of the spring box to a provision on the cross frame rather than to the column, as shown—the spring in that instance being *above* the piston, rather than *below* it, as in the present example. Other modes for disengaging are in vogue, different

* *Rule.*—To the sum of the extreme pressure (per square inch) add four times the sum of the even pressures, and twice the sum of the odd pressures; then this sum, multiplied by one-third of the distance between the consecutive points at which the pressures are taken, will give the work done expansively on one inch of the piston in one stroke.

to that shown. In some cases a hanging loop is preferred, attached to a lever, the other end of which—when in gear—lifts the connecting rod, and thus forces the roller from the cam. These deviations referred to, apply especially when cross frames are introduced, as all the fixed points of connection are secured to these details.

Fig. 49.



CAM EXPANSION GEAR FOR OSCILLATING ENGINES.

The side elevation shows the cam, lever, and motion at half-stroke—the steam pipe and valve casing being in front of the lever, and the trunnion plummer-block at the back. The sectional elevation represents the casing, valve, pipe, trunnion-block, and portion of the cylinder and lower frame in section. The expansion gear is shown complete, also the cam, but the entablature is shown half in section, and in complete elevation in the side view.

The action of the gear will be thus understood. The cam seen in the side elevation is a series of cams of unequal surfaces, or curves; this for the alteration of the grade of expansion when requisite. The motion of the cam is imparted to the throttle valve by a series of levers. These levers are cast together with one boss, thus forming a carriage, the shaft of which is secured to the entablature. The perpendicular lever connected to the spring box is always inclining towards the opposite column, and thus the roller between the angular levers is pressed on the cam; which latter, when revolving, causes the horizontal lever to rise and fall. The connection of the motion lever to that on the throttle valve spindle is by a single connecting rod. At the side of the casing is seen a perpendicular rod in connection with that on the horizontal line. This latter works in a slot in the former, and, with the use of the set screw, any position or angle for the valve can be retained. Should the cam be required to revolve without affecting the valve, the roller can be thrown out of gear by the handle, and the whole of the levers will be held in the direction indicated by the arrows, the set screw rendering their position stationary. When the grade of expansion is required to be altered, the hand-pin or spindle of the roller, seen in front of the cam in the sectional elevation, is turned to shift the position of the roller opposite the required curve. The spindle is screwed in the present example, and a set-nut outside the boss retains the required position. In other cases, recesses are turned in the spindle, and a stop

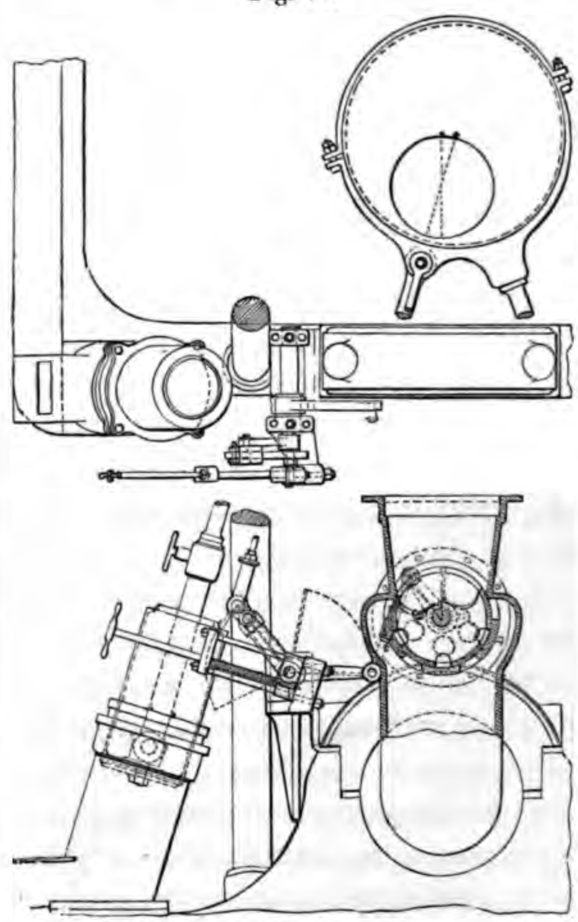
pin in the boss holds the spindle from lateral movement. The illustration is a fair specimen of the gear in question, also the entire disposition of the detail depicted is simple and effective. For a pair of engines of 250 horse power nominal collectively, the following proportions are in practice, being compiled from a late example :—

	Ft.	In.
Number of Grades on Cam 4 ($\frac{1}{6}$, $\frac{1}{3}$, $\frac{1}{2}$, $\frac{1}{12}$)		
Extreme radius of Cam	0	$9\frac{1}{2}$
Least Radius	0	$7\frac{1}{4}$
Width of Cam	0	3
Width of each Curve	0	$0\frac{3}{4}$
Diameter of Roller or Motion Wheel .	0	$7\frac{1}{2}$
Diameter of Spindle	0	$1\frac{3}{8}$
Diameter of Carriage Shaft . . .	0	$2\frac{3}{8}$
Length of Carriage	0	$7\frac{1}{2}$
Diameter of Throttle Valve . . .	1	6
Thickness of ditto	0	$0\frac{5}{8}$
Diameter of Spindle	0	$1\frac{5}{8}$

The next example which deserves attention as a simple and effective arrangement, is that generally adopted by Messrs. Maudslay, Sons, and Field: this firm prefers the disposition of the detail as illustrated by Fig. 50. The "side elevation" shows the valve and casing in section, and the gear complete behind the casing—the relative position of the gear will be better understood by referring to the plan. The eccentric, shown in elevation above the plan, is of the ordinary kind with a loop or band, having three rods connected to the lower side. One rod, it is seen, is connected to the gear in question, and the remaining two to the feed and bilge plungers—one pump being only shown in elevation and plan.

The valve is cylindrical, with openings for the admission and regulation of the flow of the steam. The passage or opening in the upper side of the valve admits the steam, and the four openings below permit its escape to the passages in the casing. The openings in the inner shell of the casing correspond with those in the valve, and the widths of the solid portions of the latter determine the laps required. After the steam has passed through

Fig. 50.



MESSRS. MAUDSLAY'S ECCENTRIC EXPANSION GEAR.

the valve, it flows into the passage in the casing, and from thence into the trunnion opening. The valve and casing are of such simple character, that the sectional view only is deemed requisite to pourtray their form in plan also.

The motion required for the valve is obviously vibratory, to admit and intercept the steam, which is attained by the rotary motion of the eccentric. The connection of

the latter to the valve spindle is by a series of levers, with separate shafts and connections. The motion from the eccentric is transmitted in the following manner. The rod attached to the band is connected to the first motion lever —depicted at an angle—a second lever being shown on the horizontal line. This last—or second—lever transmits its action to a third but shorter lever, which is keyed on the valve spindle; the connection with this lever being attained by an adjusting rod.

On alluding to the plan, it will be seen that the first lever—seen in elevation as one—is actually two levers at the same angle. The front portion is secured on a pin, and the back part separately secured on a shaft, supported by a bracket cast with the trunnion plummer block. Inside the first bearing is the second lever—horizontal—in the elevation.

Obviously, from this connection, on the eccentric causing the first lever to descend, the second lever will ascend, and the valve, and third lever, will oscillate, due to their position on the spindle.

Thus far the motion is rendered apparent. Further attention is now requisite to understand the mode of regulating the grades of expansion, to which the remaining portion of the gear—not yet alluded to—pertains. Forming a portion of the bracket alluded to, is a projection for the purpose of supporting two guide rods, angularly located, and sustained at the extremities by a cap and nuts. Clasping the rods, at the back end, is an adjusting block, the motion for which is acquired by a third rod screwed between its bearings, and prolonged beyond the cap with a cross-handle. The two levers alluded to, termed the first motion lever, are attached together by a dove-tailed portion at the extremity of the inside lever, the latter, by its connection with the shaft, remaining laterally fixed, while the former slides in common with the action of the block.

The dotted angular lines and arcs denote the travel of each lever, when the block is in the position illustrated. Now on lengthening the "first lever"—by shifting the block—the travel due to the eccentric being unalterable, the motion of the "second lever" will be lessened in proportion to the difference in the radii described by the levers—both being shown as equal in the illustration.

Further, if the motion or travel of the "second lever" is reduced by lengthening the "first lever," obviously the travel of the "third" or "valve lever" will be affected proportionately. It will be noticed, also, that the shifting of the "adjusting block" not only lengthens the lever in question, but it also shifts the valve on its axis, and thus the grade of expansion is altered. Complex as this description may seem, yet, on dwelling on its purport and consulting the drawing, the matter under notice will be readily apparent.

The illustration presented is reduced from the working drawings of the "expansion valve and gear" for the engines constructed and fitted by the firms alluded to in W. I. mail packet "Mersey," 250 horse power nominal collectively. The following dimensions convey a correct idea of the proportions adopted by the firm in question:—

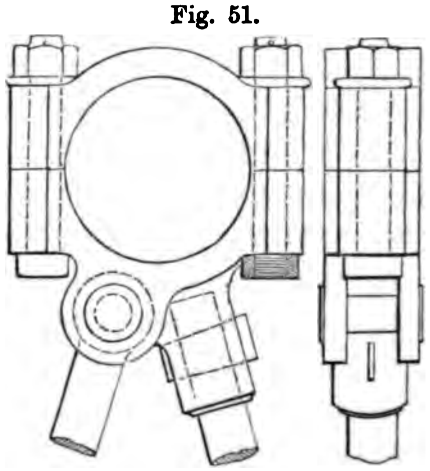
	Ft.	In.
Diameter of Expansion Valve	0	$12\frac{1}{8}$
Diameter of Steam Pipe	1	1
Width of Openings in Valve	0	2
" " Casing	0	2
Diameter of Eccentric	2	$1\frac{1}{4}$

	Ft.	In.
Diameter of Crank Shaft	0	12½
Diameter of Bolts for ditto	0	1¼
" of Expansion Rod	0	1¼
" of Plunger Rods	0	2
Diameter of First Lever Shaft	0	2
" of Valve Spindle	0	1⅞
Length of First Lever (as shown) . .	0	10¼
Length of Second Lever	0	10¼
Length of Third Lever	0	6
Stroke of Eccentric	1	1¼
Stroke of First Lever (as shown) . .	1	1¼
Travel of Second " . . .	1	1
Travel of Third " . . .	0	4⅜
Length of Sliding Space for Adjusting Block	0	8
Diameter of Trunnion Bearing	1	8½
Diameter of Pump Plunger	0	10⅜

AIR PUMP MOTION AND CONNECTIONS.

The air pumps of oscillating engines usually erive their motion from the intermediate rank shaft. To produce the requisite connecion of the shaft and the piston, two means nly at present are universal, either by cranks r eccentrics. Now with each of these details he principle of motion is alike; the crank pin escribes a circle, and also the centre of formaion of the eccentric, during the rotary motion. 'he air pumps are mostly located angularly, on ach side of the condenser, and the termination f the angles are at the centre of the shaft. 'his will be understood by referring to Fig. 45 page 201), the angles of construction being hown by dotted lines in the "sectional elevaion." Next to be considered, after determining he position of the pumps, is the length of the ods. This dimension will, in a great measure, epend on the means adopted for guiding the liding point of connecting rod. The present ractice is to use "trunks," as shown in Figs. 2, 43, and 45, in pages 196, 197, and 201, to uide the sliding point of the rod, locating the in near the piston. In some instances the in is midway of the trunk's length, in order to reduce the diameter of the trunk; but, by this, the strain on the latter is, of course, greatly increased to that imposed when the pin is lower down. It is for this cause that the connection is generally adopted, as depicted in the illustrations alluded to. It will also be noticed that when the piston is at the full up-stroke, the centre of the pin is below the gland of the stuffing box. Now these marks apply directly to the preliminary considerations common to the "air pumps" and mode of working the pistons, and the next step is the form and construction of the connecting rod. When two air pumps are used, they are either directly opposite each other in plan, as in Fig. 42—page 196—or the centres are apart equal to the width of the connecting rod bearing, plus the clearance. The arrangement illustrated requires only one connection with the crank pin. One rod is keyed into the lower portion of the brass, and the other is hung on a pin on the opposite side. The keyed rod always forms a direct line from the centre of the crank pin, but the opposite rod a broken line at the point of suspension at certain portions of the stroke. This

Fig. 51.



AIR PUMP, TWIN CONNECTING ROD HEAD.

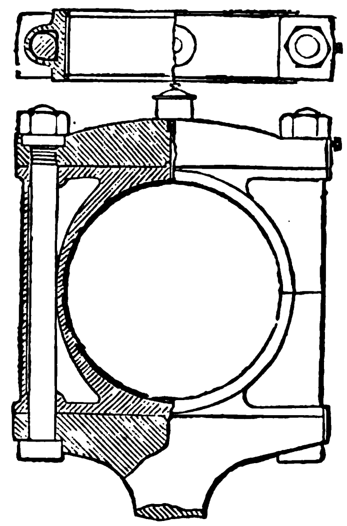
will be better understood by alluding to Fig. 51,

which is a "side and end elevation" of the crank end of the rod in question. The cap is made of wrought iron, and the lower portion of gun metal. The firms most partial to this mode of connection are Messrs. Maudslay, Sons, and Field, and Messrs. James Watt and Co. Other firms prefer to make the entire portion of wrought iron, with gun metal lining, to reduce the friction and wear with the crank pin. The trunk end of the connecting rod is secured by a single-eye, for the rod, and a double-eye bolt in the trunk, as seen in Figs. 42 and 43. The bolt is attached to the piston by a nut recessed in the body, by which means a separate provision for clearance is not required in the lower valve plate. The brass in the eye is in the lower half only, as the labour of the pump is greatest when ascending. The adjustment of the pin and brass is often attained by a cotter on the upper side of the eye; but in many instances this is omitted, as the friction is proportionately slight to that on the crank pin. The rod is in two portions, connected, about centrally of its length, by a socket and key, this being for the purpose of erecting with facility, and portability as spare gear. The following dimensions are common with those given on page 197:

	Ft.	In.
Diameter of Crank Pin	1	5
Length of Bearing	0	9
Diameter of Cap Bolts	0	3½
Distance between centres of Bolts	1	9
Thickness of Cap	0	2 13/16
Diameter of Connecting Pin	0	4
Thickness of Eye in Rod	0	4¼
Thickness of Eyes in Brass	0	2¼
Mean width of Connecting Key	0	4¼
Thickness do.	0	0⅞
Diameter of Socket	0	7
Diameter of Lower Socket in Rod	0	6½
Diameter of Trunk Eye Pin	0	3½
Diameter of Trunk Piston Pin	0	4¼
Length of Rod between centres	12	5⅛

The illustration Fig. 52 is the crank end of the connecting rod, belonging to the air pump, shown by Fig. 43, in page 197. In that example the centres of the air pumps are 2¼in. on each side of the centre of the condenser, thus making a difference of 4½ inches for the port and starboard positions.

Fig. 52



AIR PUMP, SINGLE CONNECTING ROD HEAD.

The form of the head in the present example is the more general shape than that of Fig. 51—page 220. This to be accounted for from the fact, that independent connection admits of a similar means of disconnection, which, with large engines, is no mean advantage. It is obvious, therefore, that the marine engineer has to consider weight of material, when producing compactness of arrangement; and it is from this cause, that the ideas of young engineers are often a source of vexation and trouble, when not corrected in due time.

The example under notice is shown in sectional and complete views. The cap is of wrought iron, increased in thickness at the centre, to resist the direct line of strain in-

curred during each stroke of the pump. The brasses are in two portions, the line of division being shown in the complete side of the "elevation." Each brass is hollow, to reduce the material, while at the same time the requisite strength is preserved. The T head of the rod is shaped as the cap, and both are connected to the brasses by the bolts, of ordinary description. Each nut is recessed in the cap, and set and stop pins prevent looseness possible.

This form of head and connection of the details is much used by many of the leading firms already alluded to, in connection with the air pump and condenser, by Fig. 43 (page 197). The following dimensions are in relation to those given in page 198:—

	Ft.	In.
Diameter of Crank Pin	1	2½
Length of Bearing of Brasses	0	4½
Diameter of Cap Bolts	0	2
Distance between Centres of do.	1	5½
Width of Cap and T head	0	4¼
Thickness of Cap at Centre	0	3
" " Ends	0	2
" T head	0	1¾
Thickness of Brasses at top and bottom	0	1
Thickness at centre	0	0½
Diameter of Connecting Rod at T head	0	4¼
Diameter of Eye Pin	0	3¼
Diameter of Trunk Pin	0	4¼
Length of Rod between centres	9	8⅜

In some instances it is preferred to make the head of two solid portions, with thin circular or octagon brasses. An example of this kind has been designed by ourselves for the air pump of a pair of engines of 150 nominal horse power collectively, illustrated by Fig. 44—page 199. The diameter of the crank pin is 9½ in., the cap bolt 2⅛ in., and the length of connecting rod, between centres, 5 ft. 6 in. Each nut is recessed in the upper portion of the head, and set screws ½ in. diameter prevent looseness. The brasses are ¾ in. thick at the line of strain, and ⅝ at the sides; the flanges being ¾ in. in thickness, and length of bearing 6 inches. The top portion is 2½ in. thick; and that formed with the rod the same, at the angles with the brass.

The principal gain with the solid head is that the bolts are more firmly sustained than with the loose cap and flat brasses. Messrs. J. Penn and Son prefer the latter kind, or similar, as Fig. 52—page 221—with raised portions at the connecting surfaces; but the caps are flat throughout, while in some cases the caps are dispensed with, similar to the example depicted by Fig. 51—page 220.

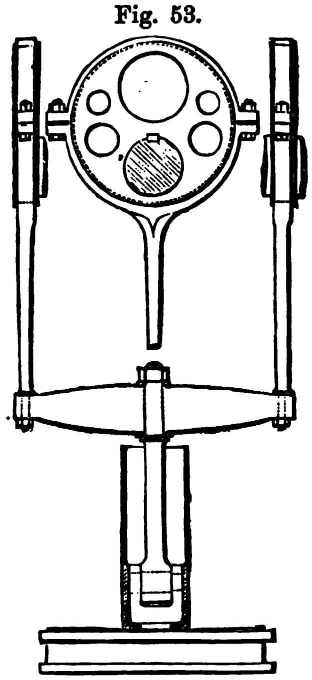
The next means of imparting motion to the air pump piston is, as before stated, by eccentrics. The firm most partial to this mode of action is Messrs. James Watt and Co., who have lately fitted no less than five vessels with diagonal oscillating engines, and eccentric motion for the air pumps. Messrs. Ravenhill also have constructed engines of similar type, and other firms in England and Scotland often adopt the same arrangements. In fact, so universal is the system, even for engines of moderate power, that the new saloon steamers on the Thames are fitted accordingly.

The motion under notice has the advantage that one or two eccentrics can be arranged to work the same pumps. Now with two eccentrics, the strain on the intermediate shaft is transmitted close to the bearings, a matter of great consideration with large engines, when the distance between the centres of the entablatures is considerable. One fact, of course, in connection with the duplicate motion,

is that a crosshead is requisite, or additional detail, but the shaft can be of less diameter than when the crank or single motion is adopted. Other facts also are apparent, such as the reduction of the labour and material for the shaft; and when the piston rods of the cylinders are connected on the same crank pin, the duplicate eccentric motion is particularly available.

The illustration, Fig. 53, is an example of twin-eccentric motion. The trunk and piston

Fig. 53.



AIR PUMP, TWIN-ECCENTRICS AND CROSSHEAD MOTION.

are of the ordinary kind, also the connection of the rod. The eccentrics are shown in end and side elevation, to render their form obvious. For an air pump 3 ft. in diameter, and a stroke of 15 in., the following dimensions are in practice:—

	Ft.	In.
Diameter of Trunk	0	11
Length of ditto	2	6¾
Distance between Centres of Pin and Crosshead	2	5½
Distance between Centres of Crosshead and Intermediate Shaft	4	0¼
Diameter of Intermediate Shaft	0	10
Diameter of Crosshead Rod	0	3½
Diameter of Eccentric Rods	0	2¾
Diameter of Eccentric	2	2¾
Diameter of Eccentric Bolts	0	1¼
Width of Eccentric	0	4¼
Width of Eccentric Band	0	3⅛
Distance between Centres of Eccentrics	3	7¼
Depth of Crosshead at Centre	0	9¼
Ditto ditto at Ends	0	4
Diameter of Boss at Centre	0	6½
Ditto ditto at Ends	0	4½
Thickness of Web at Boss	0	2½
Thickness of ditto at Eye	0	1¼

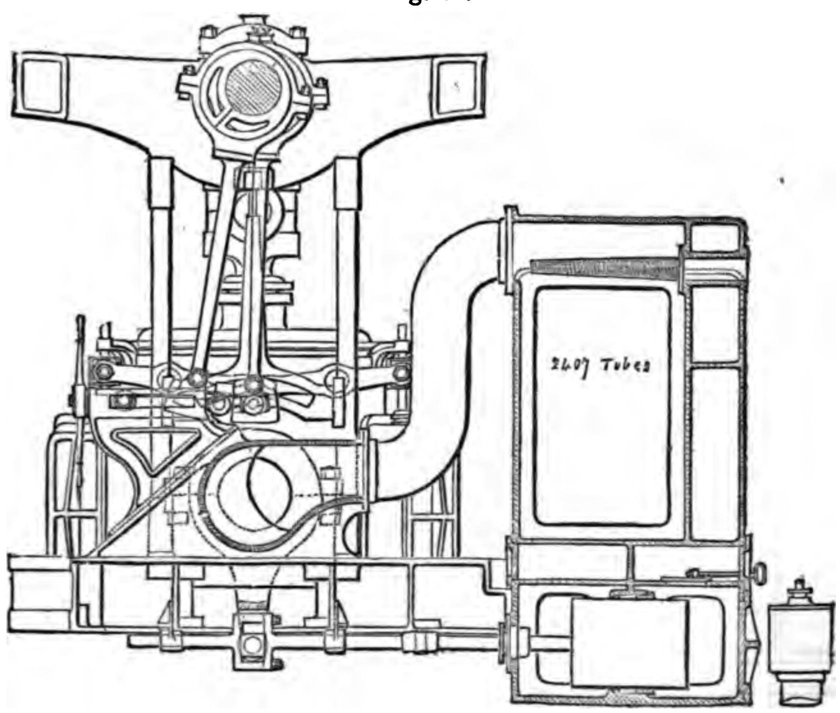
In page 203 a notice is made of a mode of horizontal motion for air pumps by Messrs. James Watt and Co.: the illustration, Fig. 54 —page 224—is a sectional elevation of the engines there alluded to. The condenser and exhaust trunnion pipe are shown in section, the remainder of the engine being in complete view. The link motion and starting gear is precisely as that depicted by Fig. 46—page 206—therefore needs no comment at present. The condenser is fitted with 2,407 tubes, each 6 feet long and ½ inch diameter. The circulating and air pumps are each 18 inches in diameter, and the stroke the same in length. The arm secured on the cylinder gudgeon imparts motion to the rod connected to each pump, and the slot added to the rod prevents the vibration curve, or versed sine, affecting the motion. The rods are guided fore and aft of the centre of motion by brackets each 4 inches long, secured to the lower frames. The arm clasps the rod by a double eye, the brasses being sustained in the slot, and the connecting pin, 3 inches in diameter, is of the usual kind. Each rod is 2½ inches in diameter, and 4 feet 5½ inches long,

from the connecting socket, the length of which is 6½ inches, and its diameter 4 inches. The length of the arm, between centres, is 2 feet 5⅛ inches, therefore the versed sine of the arc of motion = 1·4 inches. The length of the working barrel or plunger of the pump is 1 foot 11 inches, and the depth of the packing groove 4 inches. The exhaust steam pipe, 11 inches internal diameter, is connected to the top of the condenser. Opposite to which connection an injection pipe is secured, perforated all round; the diameter of the pipe at the flange is 4 inches, and at the extremity 2½ inches, the length being 2 feet 6 inches. This pipe is of course only used when the tubes are out of order, or the injection system of condensation preferred. The length of the condenser in the present view is 3 feet 9 inches, and the height 7 feet 9½ inches.

Thus far the elevation is rendered obvious, and the plan must next engage attention. This view is represented by Fig. 55—page 225. The link motion and starting gear having had prior attention, it will be sufficient to state, that instead of a wheel on the screwed shaft for manipulation, four handles at right angles are preferred in this instance. The crosshead is retained on the shaft by a set or stop screw that closes the sliding block. The entire arrangement is so simply understood from the illustrations, that the remainder of the description will be concise.

The diameter of each of the cylinders is 3 feet 7½ inches, and the length of the stroke of piston 4 feet, equal to 120 horse power nominal collectively. The distance between the centres of cylinders is 8 feet 4 inches, and that for the pumps 1 foot 10 inches. It will thus be

Fig. 54.

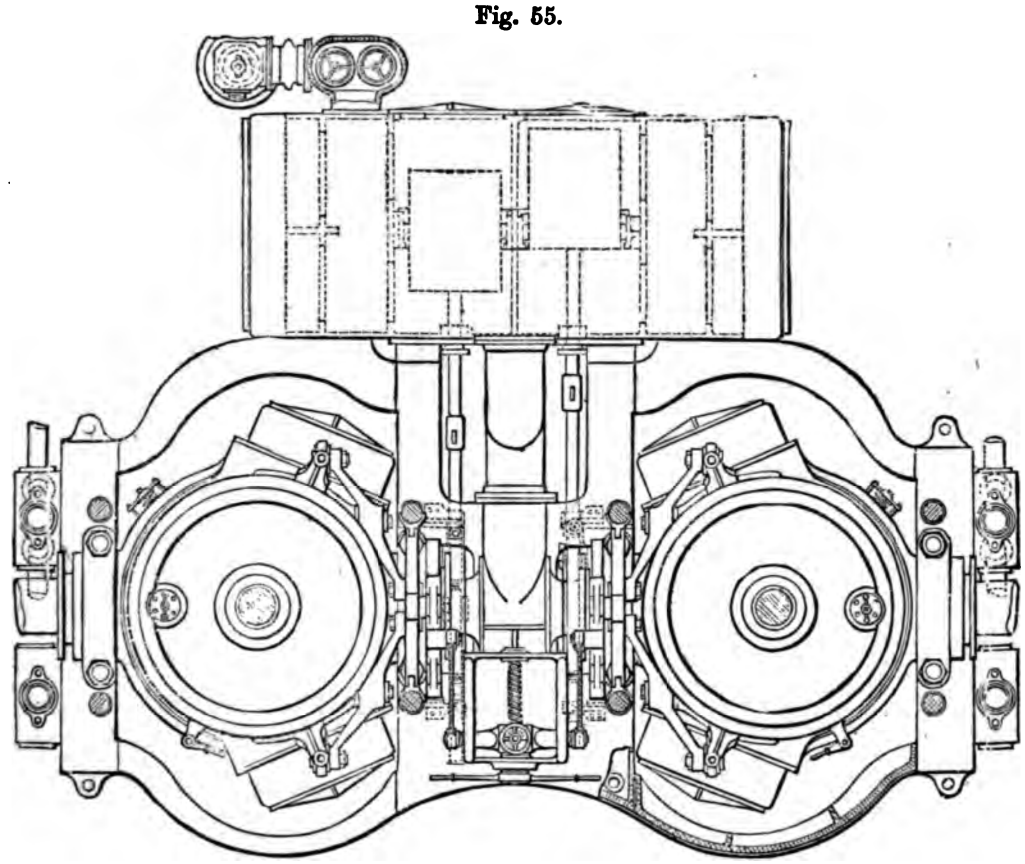


MESSRS. WATT'S OSCILLATING ENGINES WITH SURFACE CONDENSERS.

understood that the pumps are centrally situated of the width of the condenser: it is almost needless to add that only two pumps are adopted—one exhaust steam gudgeon imparting motion to the circulating pump and the other to that for draining the condenser. The suction and discharge valves are suitably placed, and doors respectively secured, for access, inspection, and repair. The extreme width of the condenser, transversely of the hull, is 8 feet 10½ inches.

This mode of motion for the pumps admits of horizontal and double action, thus retaining compactness of arrangement and the least diameter for the plungers. To fairly test these facts is to revert to Fig. 45—page 201—where the condensers are separate and the pumps single and vertical action. The arrangement now under notice and illustration is similar, with reference to the mode of motion, to that advocated in page 204, where "the centre of the bottom of the cylinder is mentioned as the best point to derive the motion for the pumps." The difference is, that the condenser in the present example is common to both engines, also the action of the pumps is dependent on the motion of both cylinders. Now when each engine works both circulating and air pumps, an independent cause and effect is certain, and in the event of either engine being disabled, the same system of condensation can be maintained for the other.

Fig. 55.



MESSRS. WATT'S OSCILLATING ENGINES WITH SURFACE CONDENSERS.

The arrangements of surface condensers for the oscillating engine are not much varied, the examples already alluded to being the general practice, therefore, further comment is needless.

CENTRIFUGAL PUMP.

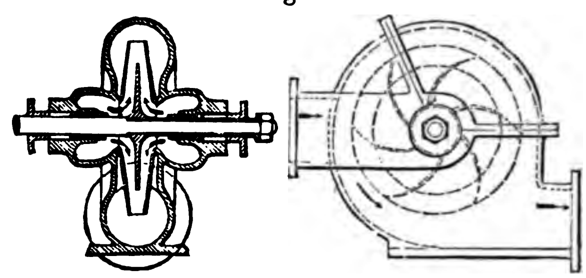
It is preferred by some marine engineers to use the "centrifugal pump," as an agent to circulate water through the condenser; driven, as before stated, by separate engines in most cases.

The authorities on the efficiency of these pumps do not arrive at a mutual conclusion on all the points connected with the same. It is argued by some that the arms should be curved at the outer ends, and by others that they should be radial at that part, while others, adverse to both of these suggestions, advocate a continuous curve as the correct form. Now, admitting that each authority claims notice, it seems on reflection that—considering the fluid as a body exposed to centrifugal force—the more readily the arms can be relieved at the outer extremities, the greater will be their efficiency. The theory held by many is, that the form of the arm should be in recognition of the speed, and that the flow of the water should be a direct line from the centre of motion.

The accompanying illustration, Fig. 56, is an example, in practice, adopted by the leading firms. The blades of the fan are curved according to the most approved form, to ensure the least loss of power.

The central or divisional plate of the fan is secured on the shaft by a key, and lateral disturbance is prevented by the long bush fastened by a nut at the shaft's extremity. The shell of the apparatus is in two portions—for the purpose of erection—connected at the flanges, seen in the side elevation, by bolts

Fig. 56.



CENTRIFUGAL PUMP.

and nuts. Stuffing boxes are provided at each side of the shell, which not only prevent leakage, but also support the shaft. The arrows indicate the flow of the water, and thus the action of the pump can be readily understood.

The type of engine is single piston rod direct acting. The piston rod is guided by a slipper guide. The cylinders—two—are cast together, and likewise the head stocks of the bearings for the crank shaft. The slide valves are at the outside of each cylinder, worked by ordinary eccentric motion.

The following are the principal dimensions of the pump and engines, which will readily cause a truthful conception of the practice at the present time:—

	Ft.	In.
Diameter of Fan	2	0
Diameter of Divisional Plate	1	6
Diameter of Shaft	0	3
Width of Casing at Centre	0	5
Width at Edge	0	2
Radius of Internal Curve of Shell	1	3
Diameter of Suction and Delivery Openings	0	9
Height of Centre of Suction Opening from Base of Shell	1	6
Extreme Length of Shell	3	5
Width of Shell at Suction Side	1	7

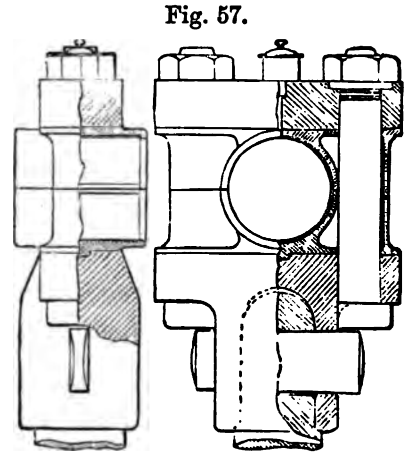
	Ft.	In.
Width of Shell at Delivery Side	1	3
Diameter of Steam Cylinders	0	8
Length of Stroke of Piston	0	8
Width of Supply Port	0	0¾
Width of Exhaust Port	0	1¼
Length of Ports	0	4
Length of Stroke of Slide Valve	0	2¼
Diameter of Piston Rod	0	1¼
Diameter of Slide Valve Rod	0	0¾
Length of Connecting Rod	1	4
Diameter of Block Pin	0	1⅛
Thickness of metal of Shell of Pump	0	0¾
Ditto ditto of Steam Cylinders	0	0½

STEAM CYLINDER PISTON ROD MOTION, AND CONNECTIONS.

The details in connection with the piston rod, although similar in principle to those for the air pump, are entirely different in their application, and therefore in proportion. The cause for this is that with the air pump the piston connecting rod is the prime mover, whereas with the cylinder the piston imparts the motion.

On further entering into this question, it will be remembered that the steam is the main agent, and has to overcome the load on the pump. It will be apparent therefore from these facts, that the steam has to be considered when determining the proportions of the detail under notice, as far as the direct line of strain is concerned, but of course the lateral resistance must not be overlooked. The strain on the piston rod will be the greatest when the cylinder is at its utmost angle, but if the latter is correctly balanced, the lateral strain will be lessened and friction also reduced: and it is partially to sustain the rod from deflection or bending, that the deep bushes are generally inserted in the lower portions of the stuffing boxes. It will be remembered in the proportions of the rods in relation to Figs. 39 and 40 —pages 194 and 195—that the former is 9 inches in diameter, and the latter 7¼ inches, the diameters of the cylinders being respectively 64 inches and 61 inches. Very simple calculation only is requisite to prove the difference in the relative proportions of the rods to the areas of the cylinders, this being duly recognised by the difference in the depths of the bottom bushes.

Fig. 57.



CYLINDER PISTON ROD HEAD.

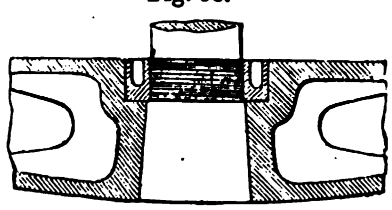
The design of the head of the rod that is most universal, is as that shown above by Fig. 57, in half sectional and complete elevations. The cap is flat, and the brasses of similar design, where in contact, for stability. The lower portion of the head is a T socket, into which the rod is secured by a key. In the end view it will be noticed the diameter of the socket and width of the cap are unequal: this, however, is not a general proportion, it being preferred by some makers to retain equal dimensions, and introduce a circular flange to the brasses to prevent lateral disturbance. This latter mode is the design of the head in common with the cylinder illustrated by Fig. 39 —page 194—and the following are the dimen-

sions of the head in question also for that of Fig. 57.

	Fig. 39. Ft.	Fig. 39. In.	Fig. 57. Ft.	Fig. 57. In.
Diameter of Crank Pin	0	11	0	8½
Diametor of Piston Rod	0	9	0	7¼
Diameter of Bolts	0	5¾	0	3¾
Distance between Centres of Bolts	1	6	1	2
Thickness of Cap	0	5	0	3¾
Width of Ditto	0	10	0	6½
Thickness of T portion	0	5	0	4
Width of ditto	0	10	0	6½
Diamoter of Socket	0	11	0	10
Length of Socket	1	4	0	11¾
Mean Width of Key	0	6¾	0	4 7/16
Thickness of ditto	0	2	0	1½
Thickness of Brass at the Cap	0	1½	0	1⅜
Ditto of ditto at T portion	0	1¼	0	1⅜
Length of Bearing	1	0⅞	0	11¼

The mode of securing the rod in the piston is usually by a nut, either on the upper or lower surface. When the nut is at the under side it often projects, and therefore a recessed provision is requisite in the cylinder bottom. It is for this reason that the upper side is often preferred with the nut recessed in the piston.

Fig. 58.



MODE OF SECURING ROD WITH PISTON.

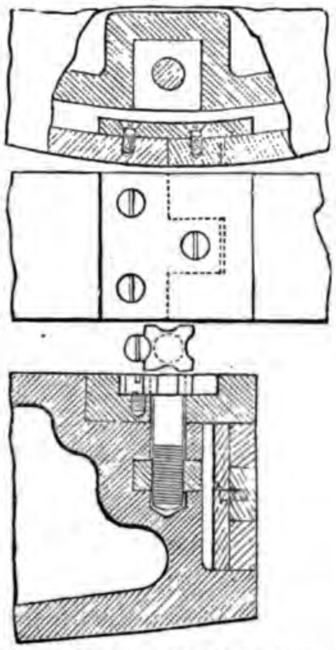
The section of the piston and mode of securing the rod are illustrated by Fig. 58, being the universal practice by the most experienced makers. The nut has four or more holes drilled in the top side, for the purpose of turning the same, and securing the rod in its bearing. The rod is tapered in the piston in this case, while in others a flange or collar is preferred at the extremity, recessed in the piston: that for Fig. 39—page 194—is thus shaped. The dimensions of the illustrated example, and the one descriptively alluded to, are as follows:—

	Fig. 39. In.	Fig 58. In.
Diamotor of Screw	9½	8
" Nut	14	12
Depth of Nut	3½	3½
Diameter of end of Tapered Portion		9⅞
Depth of Piston	12	11½
Diameter of Collar	12⅛	
Thickness of ditto	3⅜	

The remaining portion of the piston requiring especial attention is the packing ring. This detail is of the greatest importance, as the existence of any defects in it lowers the power of the engine. The main effect to be attained is an uniform contact of the ring with the internal surface of the cylinder, so that no steam passes the piston. To accomplish this the spring ring has had a fair trial. This ring is divided either angularly with a tongue piece, or slotted with a back plate. The sections shown by Fig. 59—page 229—is the latter mode of preventing the steam passing the ring at the division. The ring is sustained against the cylinder by packing behind it, a gasket of hemp being often used; while, with other examples, steel springs are preferred, similar to the piston illustrated by Plate 11—which if superheated steam is used is the better plan. The mode of retaining the ring is by a face ring secured on the top side of the piston, by studs screwed in gun metal nuts dovetailed in the body of the piston. To prevent looseness of the studs, each has a *set* stud at the side of the collar, the plan of both being shown directly over the elevation—the recess in the face ring being sufficiently large to admit the use of a box spanner. The

cylinder, in connection with the portion of the piston and rod, illustrated by Figs. 57 and 58

Fig. 59.



PISTON SPRING RING AND STUDS.

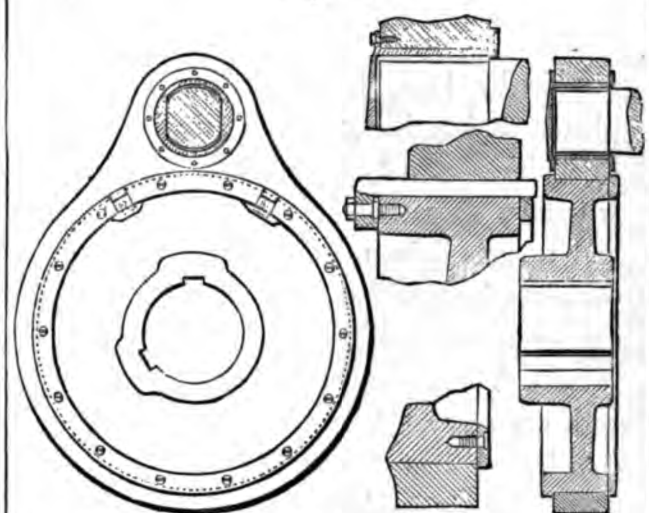
—pages 227 and 228—is 61 inches diameter, and the remaining dimensions are as given below :—

	In.
Thickness of Metal around Rod	2
Number of Ribs cast with Boss, 6.	
Thickness of ditto	$1\frac{1}{8}$
Width of Ribs from Boss	6
Thickness of body of Piston	$1\frac{1}{4}$
Width of Face Ring	$5\frac{1}{2}$
Thickness of ditto	$1\frac{1}{2}$
Diameter of Securing Studs	1
" Set Studs	$0\frac{3}{8}$
Number of Studs, 10.	
Side of square of Nuts	$2\frac{1}{4}$
Thickness of Metal around Nuts	$0\frac{7}{8}$
Thickness of Nuts	1
Thickness of projection under Packing Ring	1
Width of Ring	$4\frac{3}{4}$
Thickness of Ring	$0\frac{7}{8}$
Space for Packing	$0\frac{7}{8}$
Thickness of metal beyond Packing space	1

CRANK DISCONNECTION GEAR.

In the event of the engines becoming disabled, or a similar occurrence happening to the paddle wheels—when it may be required to have recourse to sails only as a means of propulsion—the wheels must be freed from the engines, and to attain this, the cranks must be disconnected. Now, with engines of large power, or even moderate size, the withdrawal of the crank pin is a laborious as well as tedious task—to say nothing of the probability of loosening the crank during the operation. It is with the view of saving time and trouble, that the arrangement termed disconnecting

Fig. 60.



PADDLE SHAFT DISCONNECTING RING AND DISC.

ring and disc, illustrated by Fig. 60, is often adopted. This example is the general practice of Messrs. J. Penn and Son, Messrs. J. and G. Rennie, Messrs. Ravenhill and Hodgson, and other firms of distinction. The practice of Messrs. Maudslay, Sons, and Field, is to shift the paddle shaft, by gearing and a socket as a coupling, which produces the same effect.

To return to the illustrated example, the disc is of cast iron, secured on the shaft by two keys at right angles with each other. The ring is of one forging, and bored to fit the

turned periphery of the disc. It will be noticed that a projection is formed with the disc on one side, and a wrought iron separate portion secured on the other, by which means the ring is retained on its bearing—an enlarged section of the connection being shown between the elevations. The crank pin—flattened at the sides—is held in its bearing by a separate brass bush, having a flange and eight screw pins to secure the same to the ring. At the opposite side a separate ring is similarly secured: an enlarged section of this is also shown between the elevations.

The means adopted to *fix* the ring on the disc is by keys, located below the crank pin; each key being retained in its bearing by studs—shown by the enlarged section. To withdraw these keys, the nuts are removed, and wedges driven in under the gib ends cause the keys to loosen, when their removal can be readily effected. The ring can then revolve independently of the disc, and thus both are separated. It is obvious from this de cription, that the cranks are disengaged simply by the withdrawal of the disc keys, —the connection of the cranks by their insertion; and such is the efficiency of this mode, that it has become universal.

The dimensions of the details under notice being worthy of due attention, the following compilation is given from a late example in actual practice:—

	Ft.	In.
Radius of Crank Pin Circle	2	3
Diameter of Crank Pin	0	8½
Diameter of Crank Shaft	1	0½
Thickness of Boss for Shaft	0	2¾
Length of Boss	0	11½
Diameter of Disc	3	2¾
Width of ditto	0	7
Depth of Projection	0	0⅝
Width of Ring	0	2⅝
Radius of Boss for Crank Pin	0	8
Radius of Connecting Curve	3	0
Diameter of Bush	0	9½
Width of Shaft Keys	0	2½
Thickness of ditto	0	1¾
Width of Disk Keys	0	2
Thickness of ditto	0	1
Diameter of Bush Pins	0	0½
Diameter of Ring Pins	0	0¾

As the crank is in immediate connection with the above, the following is added:—

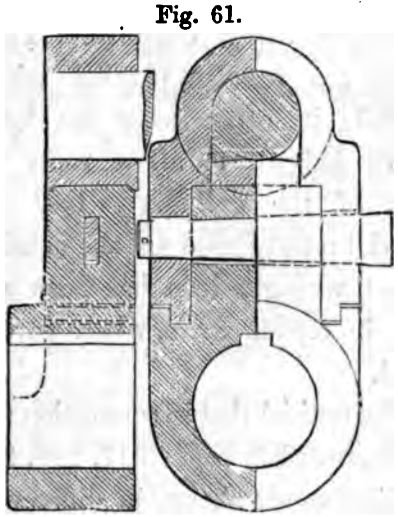
	Ft.	In.
Diameter of Crank Boss	1	7½
Width of ditto	0	11½
Diameter of Pin Boss	1	2½
Width of ditto	0	9½
Maximum width of Web	1	2
Minimum ditto	0	10⅜
Thickness of Web	0	6¾
Width of Pin Key (across bolts)	0	2¾
Thickness of ditto	0	1½

These dimensions, and the illustration of the disconnecting ring and disc, are in common with the piston rod and piston, already alluded to in pages 227, 228, and 229.

In some instances the crank pin is separated and connected by a link or loop, and in such cases, when disengaging is requisite, one of the crank pins must be withdrawn. To obviate this latter cause, we have designed a connection, illustrated by Fig. 61—page 231. With this example, the connection is attained by a strap, gib, and cotter, or if not alike them in general design, the principle is the same. The disengaging of the detail is effected by first knocking out the key and withdrawing the strap. Secondly, the paddle-shaft crank is allowed to pass from the loose or intervening portion between it and the pin. Thirdly, the removal of the portion mentioned permits the crank pin to be entirely free from the remaining portion of the crank.

Now, admitting that the details to be handled, in the present instance, is more than

Fig. 61.



BURGH'S DISCONNECTING CRANK.

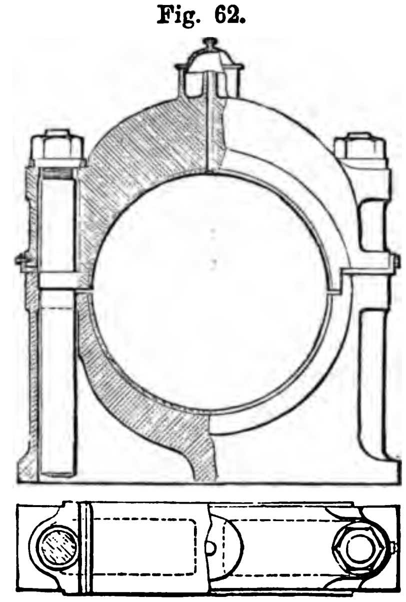
that requisite with the ring and disc arrangement, there can be no question as to the compactness, and also the economy in material and construction. The illustration under direct notice is compiled from the drawings connected with the condenser, illustrated by Fig. 44—page 199—and the principal dimensions are thus:—

	Ft.	In.
Diameter of Shaft	0	$9\frac{3}{4}$
Diameter of Boss	1	$4\frac{1}{2}$
Length of ditto	0	$9\frac{3}{4}$
Diameter of Pin	0	$7\frac{1}{8}$
Diameter of Boss	1	$0\frac{1}{2}$
Width of Strap	0	$7\frac{1}{8}$
Thickness of ditto	0	3
Mean Width of Key	0	$3\frac{3}{4}$
Thickness of ditto	0	$1\frac{1}{2}$
Lap of Strap	0	$1\frac{1}{4}$

TRUNNION PLUMMER BLOCK.

This detail supports the cylinder, and therefore is of general design. The illustration, Fig. 62, is an ordinary example. The lower portion is of cast iron, also the cap, and both are lined with gun metal strips for the reception of the trunnion bearing. It is preferred in some cases to use brasses with flanges and wrought iron caps. The cap bolts pass through the block to the lower frame, for the purpose of securing a connection without additional detail. Especial attention also has been devoted to the distribution of the metal; the cap is curved, and being exposed to an alternate direct thrust, the thicker metal is therefore introduced at the centre. Each nut

Fig. 62.



CYLINDER TRUNNION PLUMMER BLOCK.

is fitted with a stop ring and stud, and the bolts have collars on them, at the connection of the cap with the block. To prevent the cap being screwed down unevenly or too hard on the bearing, loose divisional portions are inserted at the connections, and secure from lateral displacement by studs—seen in the elevation.

On each side of the sole of the block, keys are introduced, acting between the block and the lugs on the foundation frame, and thus lateral adjustment is readily effected. The

dotted lines in the plan show the hollow portion in the block, the width of which permits a fitting strip for the sole, of the requisite area.

This example is similar to the practice of Messrs. J. Penn and Son, Messrs. Ravenhill and Hodgson, Messrs. J. and G. Rennie; the practice by Messrs. Maudslay, Sons, and Field, being depicted in Fig. 50—page 218. The example now under notice is an exact copy of that lately fitted in a despatch ship, the cylinders being 61 inches diameter, and stroke of piston 4 feet 6 inches, also referred to in page 194, the principal dimensions being thus:—

	Ft.	In.
Diameter of Trunnion Bearing	1	7½
Thickness of Lining	0	0⅝
Length of Bearing	0	7½
Diameter of Cap Bolts	0	2¾
Distance between centres of Cap Bolts	2	1½
Thickness of Cap	0	5¾
Thickness of Sole	0	1¾
Length of Sole	2	8½
Width of Sole	0	7

With reference to the lower, or foundation frame, much depends on the taste of the designer, as far as the general outline is concerned. The main considerations are the provisions for the reception of the trunnion plummer blocks, columns, cross frames, feed and bilge pumps, holding down bolts, and contingent matters. The designer has to consider also the space requisite between the frames for the oscillation of the cylinder, and to ascertain this, the greatest angle of the latter must be noticed, before settling the width. The frame is generally in two portions, connected to the condenser by bolts, studs, and nuts. The portions supporting the plummer blocks are cast hollow generally—in section as a box-girder—and the form of the section of those at the sides—connected to the condenser—as the letter ˥, the projection being outward. The ends and sides are usually parallel with each other, while in other cases the latter are curved, or bulged, to allow room for the vibration of the cylinder. Cast iron is the material mostly used for the detail under notice, but wrought iron has been adopted, but only in certain cases, where particularly stipulated.

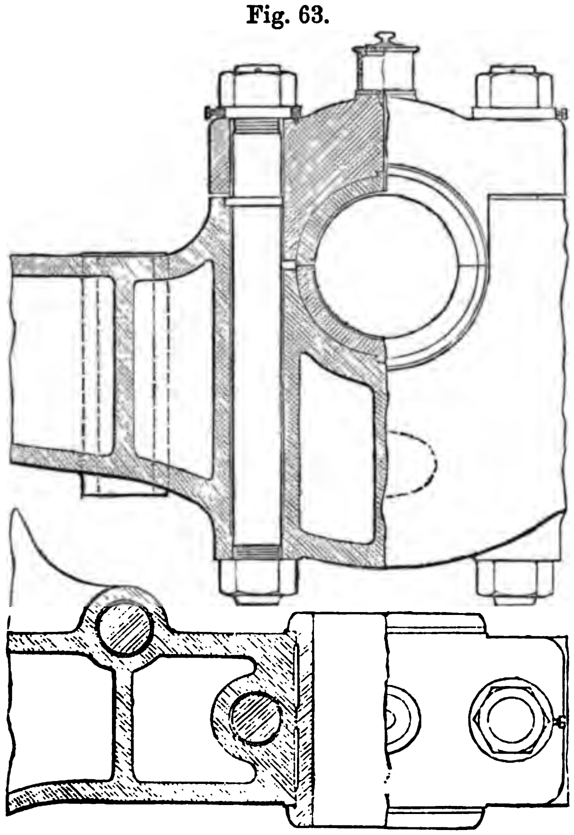
It will be noticed that in connection with the details illustrated in pages 217, 218, 225, and 234, the frames are shown in different views. If there be any preference as to design, the parallel framing is perhaps the best.

ENTABLATURE.

As this portion of the engine supports the shafting, it is obvious that strength must be preserved, even if design is sacrificed. The elevation and plan of a portion of an entablature is shown by Fig. 63—page 233—being an example similar to the practice of Messrs. Ravenhill and Hodgson. The cap, it is seen, is curved, while in other examples by some makers a flat cap is preferred, of wrought iron, with clip ends, or curved, as in Fig. 45—page 201. The portion of the detail directly under the shaft has a central rib, but in other instances this is dispensed with, and a thicker seat—than that illustrated—for the lower brass. The cap bolts are secured by a nut at each extremity—rather than a key passing through the body—at the outside of the bearing, seen in plan; but the central position should be retained if possible. The brasses are circular

—for the best means of fitting—and lugs at the top and bottom sides render them stationary. The dotted curved portion under the bearing in the entablature is the "core hole," that is stopped by a separate plate.

Fig. 63.



SHAFT BEARING IN ENTABLATURE.

Fore and aft of the bearings the entablature is usually solid, as a parallelogram in section, but a section as the letter ┐ inverted is sometimes preferred.

To economise patterns and retain portability for fitting and erection, the entablature is connected centrally—longitudinally of the hull —by flanges, bolts, and nuts, and thus a better casting can be also ensured rather than if the whole portion was entire.

To retain the rigidity of the detail in question, cross frames are often used independently of the columns. The addition, however, may be said to be a matter of idea, rather essentially, inasmuch that, if the columns are of sufficient strength and correctly situated, the cross frames will not be requisite as a means of support.

The lateral adjustment of the entablature is often effected by wedges, each having a screwed end and nut to ensure the most minute movement if required; and it is almost needless to add that these wedges are fore and aft, and generally opposite each bearing.

The example illustrated being of importance in connection with other details, the following dimensions are in common with those given in page 230, referring to the disconnection of the cranks :—

	Ft.	In.
Diameter of Main Bearing	1	0½
Length	1	6½
Thickness of Brasses	0	1½
Thickness of Cap	0	6
Width of Cap	1	3
Thickness of Metal under Brass . .	0	2⅛
Thickness of Metal of Slides of Frame .	0	3
Depth of ditto	1	4
Depth under Main Bearing from Centre .	2	2⅛
Diameter of Cap Bolts	0	4½
Distance between Centres of ditto . .	1	9½
Diameter of Columns	0	4¼
Distance between Centres of Columns .	3	10
Half length of Entablature from Centre of Cylinder	5	3

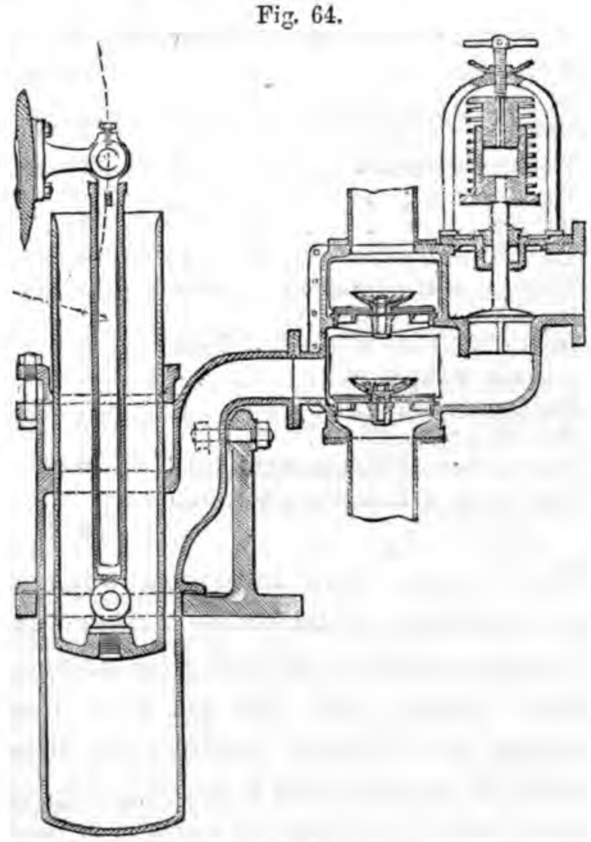
Some makers have introduced wrought iron entablatures for the double purpose of reducing the weight and increasing the strength. Messrs. James Watt and Co. have used wrought iron in many instances for these details, the form of which is shown by Fig. 46 —page 206—also a cast iron example in page 224—Fig. 54. It will be noticed that the base of the entablature forms a straight line in one instance, rather than a curve as shown

by Fig. 63—page 233—also similarly shown in page 217. The remainder of the details being nearly alike, the previous description applies to each example.

FEED AND BILGE PUMPS.

The supplying of the boilers with feed water, and the emptying the bilges of a steamer, form the main attention of the engineer when at sea, or his vessel under steam. Indeed, so far is this certain, that the air pump is generally provided with a means of draining the bilge, termed the bilge injection cock; and more than one firm has lately connected the air pump barrel and the boiler by pipes, valves, &c.

Fig. 64.



FEED OR BILGE PUMP.

The pumps under notice are mostly vertical single acting, with hollow plungers, of which Fig. 64 is a practical example. The barrel of the pump, it is seen, is recessed into the bed of the foundation frame, and connected at the branch pipe by bolts and nuts to the upper portion. The valve box overhangs the frame, and the suction and discharge valves are directly over each other—the door for access being at the side. The relief valve is directly beyond the main passage, and the spring above is of the ordinary kind, fitted with a regulating screw and guides. It is scarcely necessary to add that the suction pipe is at the bottom and the discharge at the top of the valve box, therefore an almost direct flow is the result. The origin of this arrangement is perhaps as much due to Messrs. Ravenhill and Hodgson as any firm interested in the matter under notice.

The following are the principal dimensions of the example now alluded to, lately fitted with a pair of engines of 250 nominal horse power collectively:—

	Ft.	In.
No. of Feed Pumps, 2.		
No. of Bilge ditto, 2.		
Length of Stroke of Plunger	1	8
Diameter of ditto	0	$7\frac{1}{4}$
Ditto of Barrel	0	$8\frac{3}{4}$
Thickness of Plunger	0	$0\frac{3}{8}$
Ditto of Barrel	0	$0\frac{7}{16}$
Diameter of Connecting Rod . . .	0	$1\frac{1}{4}$
Length of ditto	2	5
Width of Key	0	$1\frac{1}{4}$
Thickness of ditto	0	$0\frac{1}{4}$
Diameter of Section and Delivery Valves .	0	$6\frac{1}{4}$
Ditto of Escape Relief Valve . . .	0	$4\frac{1}{4}$
Diameter of Suction and Delivery Pipes .	0	4

The bilge pumps have no escape valves, as the discharge opening is rarely, if ever, checked or stopped.

In some instances it is preferred to place each valve in separate boxes, on each side of the barrel of the pump—the relative vertical positions being of course as those illustrated.

The remaining portion lastly to be considered is, the connection of the plunger and the mode of motion. Two universal means for imparting the action are in practice at present, either by the vibration of the cylinder, or the oscillation of the trunnion.

The illustration first under notice illustrates the plunger connected, by hollow and solid rods, to an arm secured to the steam belt on the cylinder. A key is inserted in the rod directly above the plunger, to preserve the motion, and on its withdrawal, the plunger ceases to act.

By Fig. 55—page 225—an arrangement of the pumps last under notice is shown. The valves are on each side of the plunger, and the motion for the latter is derived from an arm keyed on the trunnion—the arms being duplicate—on each side the trunnion, a reverse action for the pumps is the result.

The modes of motion thus far alluded to are adopted by Messrs. J. Penn and Son, Messrs. James Watt and Co., Messrs. Ravenhill and Hodgson, Messrs. J. and G. Rennie, and other firms of eminence.

Apart from these, the Messrs. Maudslay, Sons, and Field claim notice, in relation to the means of working the pumps under notice.

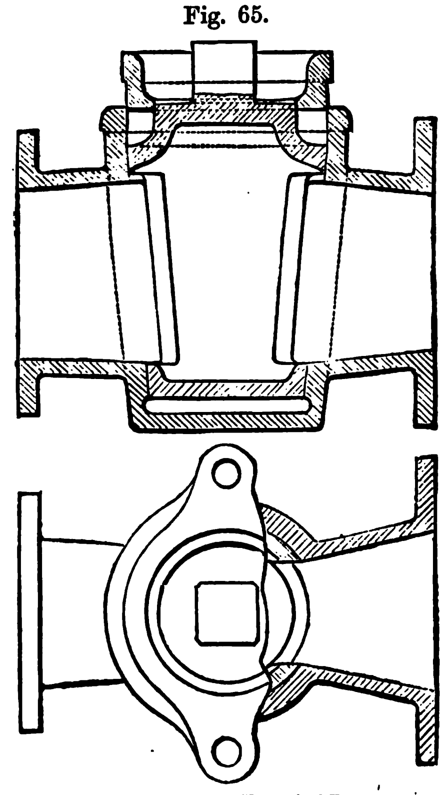
In page 218 an illustration is given of an expansion gear, the eccentric of which imparts motion also to the feed and bilge pumps. One pump only is represented, angularly secured, at the side of the trunnion plummer block. The connection of the plunger is as that for Fig. 64—page 234—but a set screw is used, in the place of a key, as a means of release. The valve box is bolted to the flange at the *base* of the pump, which situation is not as available for perfection of action as the *top* of the barrel. It needs no great enquiry as to this fact, and more particularly so, when it is remembered that *air* naturally *ascends*, and when *compressed*, with a velocity due to the pressure occasioned. It can thus be readily understood that the proper place for the discharge valve of *any* pump is *above* the *plunger*.

SUPPLEMENTARY VALVES.

INJECTION VALVE.

The detail first to be alluded to as the most important is the injection valve. The utility of this addition to the condenser is, of course, as well understood, to regulate the amount of water requisite to condense the steam. These valves are mostly of two kinds, generally

Fig. 65.



INJECTION PLUG VALVE.

either a gridiron or a plug—more often the latter—to which especial reference is made

by the illustration, Fig. 65. This is a plan and elevation of an ordinary stop cock, with direct passages, the plug being secured by a gland and packing at the larger end, rather than a screw and nut at the smaller. The solid portions of the plug prevent the flow of the fluid, and thus the position of the same, in relation to the openings in the casing, determines the admission of the water into the condenser.

When the gridiron valve is used, the movement is due to the width of the opening and lap of the valve, but with the plug valve the length of the lever greatly determines the motion to open and close the passages.

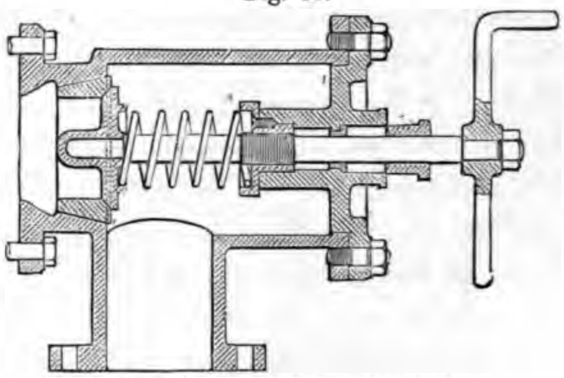
In some instances a disc valve is adopted, raised and lowered by a screwed spindle, while in other cases a lever imparts the required motion.

Irrespective of the means to regulate the flow of the water, the means of distribution within the condenser demands consideration. As due attention has been given to this, both illustratively and descriptively, in pages 196 and 197, it will be sufficient in the present instance to add that the best means of distributing the condensing water is a perforated pipe.

SNIFTING VALVES.

These details are essential when it is requisite to blow out the condenser by steam, thus drive out the air, and by condensation cause a vacuum. It is evident also when the latter function is formed, the engine can be started more readily than before, and thus the valve under notice is of utility to exclude the air as well as permit its escape. The ordinary snifting valve is vertically situated at or near the level of the bottom of the condenser, while in the example represented by Fig. 66, a horizontal position is preferred. This is an improved arrangement by ourselves, the effect gained being, that by the position of

Fig. 66.



BURGH'S SPRING SNIFTING VALVE.

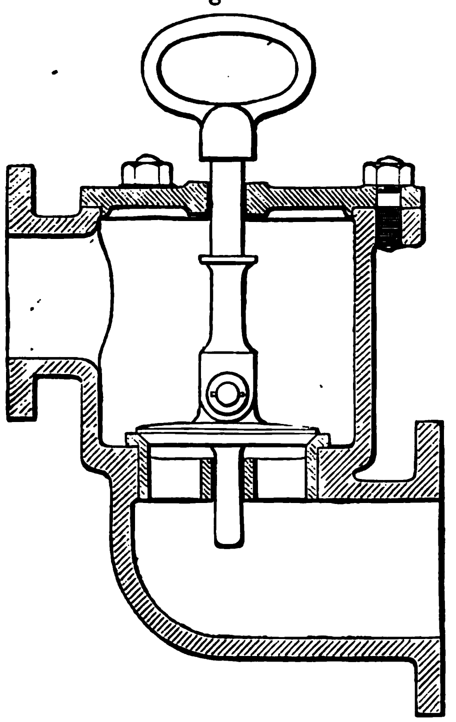
the valve, the condenser can be perfectly blown out, and the susceptibility of the valve, to the vacuum caused, is rendered more certain by the slight spring at the back. The spindle being screwed in its bearing, releases and secures the valve from and against its seat, and the portion inserted in the valve acts as a guide and stop at the same time; a sufficient clearance being allowed to prevent the valve sticking on the guiding part.

Some firms prefer the type illustrated by Fig. 67—page 237—sometimes termed the lower blow valve. In this case the valve is vertically situated above the bottom of the condenser, unless a lower provision in the latter provided. The valve is the disc kind formed with a spindle, as a guide, fitted into an eye, cast with the seating. Above the valve, a rod and handle is connected, and the valve is lifted, independently of the steam and water, by the hand.

Other examples have no casings, but simply a valve,. seating, weight, and handle; while a fourth is an ordinary stop valve and lever.

Each valve of course has to perform, by different modes, the same action or effect. Now with the example shown by Fig. 66—page 236—mechanical action is certain when the valve is released, and with that by Fig. 67, below, self action is caused; but should the valve hang or leak, the vacuum will be impaired, although space above the valve is allowed for the accumulation of water to render the valve air tight.

Fig. 67.

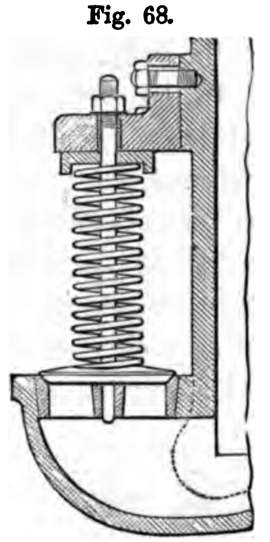


SELF-ACTING SNIFTING VALVE.

CYLINDER RELIEF VALVES.

When water accumulates in the cylinder a certain space is occupied by the fluid, and when the piston is at the full stroke, a compression of the volume must ensue. Now, apart from considering whether the water enters the cylinder either from the priming of the boilers or any other cause, immediate attention is now directed to the best means of meeting the evil alluded to. As the piston drives the water in front of its travel, it is obvious that the position of the valves in question must be at the extremities of the

Fig. 68.



CYLINDER SPRING RELIEF VALVE.

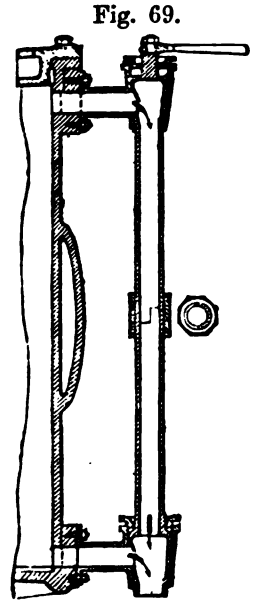
cylinder. The above illustration, Fig. 68, is an ordinary example of a relief valve, spring, and seating, often fitted, and provided for, on cylinders of oscillating engines. The nut on the upper extremity of the spindle relieves the valve from the effect of the spring if requisite. The positions of the illustrated details as at the bottom end of the cylinder, and a reverse position will be that for the top end. The spring is of course always acting on the valve, and. the latter rises only when the pressure in the cylinder overcomes that of the spring. The valve is the disc kind, guided below the seating by a spindle, and above by a bracket secured to the cylinder by three studs and nuts. The bracket also receives the

cap of the spring, while in other instances it forms part of the same.

Relief valves of another design are shown, located on the top and sides of the cylinder, by Fig. 55—page 225. In that instance each spring is retained by a cross bar and side rods; the valves—of the disc kind—are sustained as near the facings on the cover and cylinder as practicable, thus dispensing with the projections cast with the body, as shown by Fig. 68—page 237. In the event of instantaneous relief, plug valves are often introduced at the lower ends of the cylinders, and the plugs are connected with handles secured at the top end, also shown by Fig. 55—page 225.

The following illustration, Fig. 69, is the representation of a connection with the top and bottom of the cylinder, the lower plug

Fig. 69.



CYLINDER RELIEF PLUG VALVE.

forming an exit from either locality. The top plug is surmounted by a handle, and thus any motion imparted to it is transmitted direct to the lower plug. To ensure this, the connection of the plugs is a subdivision, and thus a certain action is rendered by the mode of contact. It will be noticed that the plugs are shown both open, but should a contrary effect be required, it can be readily effected by reversing their positions at the connection. This mode has been adopted by ourselves—and doubtless by others—with much success.

The remaining details of the type of engine under notice consist of matters of circumstance rather than general design. For instance, the injection valve gear may be of a given arrangement in one example and totally different in the next. The same can be stated of the gear for regulating the admission of the steam, usually termed the throttle valve gear. The information respecting the hand pump, water discharge valve, donkey engine, and other connections, being as those for screw engines, attention respecting the same to that section is directed.

PADDLE WHEELS.

The means of propelling vessels by the sliding action of a flat substance intercepting the water, is the original mode adopted by the most uncivilized race of mankind. The Indian, with his paddle, performs feats of dexterity, not only in speed, but also in directing his frail bark across the most intricate and dangerous routes. It is known also, that, before the use of machinery, the rowers of a ship formed an important mass of the crew. Now, simple as it may seem to row, or handle a paddle, there is, however, much skill to be exercised in producing the greatest effect from the least cause. Evi-

dence of this is an annual diversion with the selected members of the rival Universities of Oxford and Cambridge, and no little credit is due to the candidates on both sides, for the scientific mode with which they produce a certain speed, with the least possible area of paddle and loss of slip.

For ships of large tonnage and displacement, of course manual power is almost useless, hence the attention of the engineer is directed to the best means by mechanical effect. Before describing the most practical arrangements of the details under notice, a brief description of the principles is obviously essential.

The action of a float on entering the water is not only to displace the fluid proportionately to its cubic contents, but also to disturb the surrounding volume. The speed of the wheel and the ship should of course be nearly alike, to produce the greatest effect. The pitch of the floats should be in proportion to their width, and speed of the wheel.

For example, if the floats are too close to each other, the water will be agitated continuously, and thus a *channel* formed by the float, rather than each float acting in *dead* water. The float first entering the water should be fully immersed before the next reaches the element; and the number, acting below the surface simultaneously, must be in direct ratio to the displacement of the ship or power requisite.

Now, next to be noticed is the correct angles for the float, when entering and leaving the water. It will be remembered it is stated in page 7, that "the floats, on leaving the water, lift a certain portion of the same, but the greater amount of friction is directly after the floats pass the centre, or vertical line."

To clearly define the matter in question, presume a side view of a steamer, the paddles to be in motion. Each float on entering should descend perpendicularly, and remain in that position until reaching the centre of immersion, when the float should assume the horizontal line, and leave the water at an angle due to the speed of ascent. To produce this uneven action, a mechanical arrangement connected to each float, worked by cams, is requisite; but as this additional detail is not only complicated, but also liable to get disabled in a sea passage, the more simple means, with an uniform motion, is sometimes adopted. Wheels with fixed floats do not lose so much effect, however, as might be imagined, and this is more apparent when the floats are narrow in proportion to their area.

The *positive* slip of the paddle wheel is known by the difference in the speed of the floats and the vessel. Now, if there is no slip, each float will propel the vessel the length of the *arc* of immersion, during the time occupied in entering and leaving the water. The result is very good, however, when the distance propelled equals the *chord* of immersion during the exertion of the floats; and, indeed, this latter attainment is the more practical effect. In the event of *negative* slip, the vessel moves a greater distance during a complete revolution of the wheel than the circumference of the circle of immersion. The action of the floats is therefore checked, and their power reduced.

It is obvious from these remarks that the correct proportions of the paddle wheel form an important portion of the subjects under

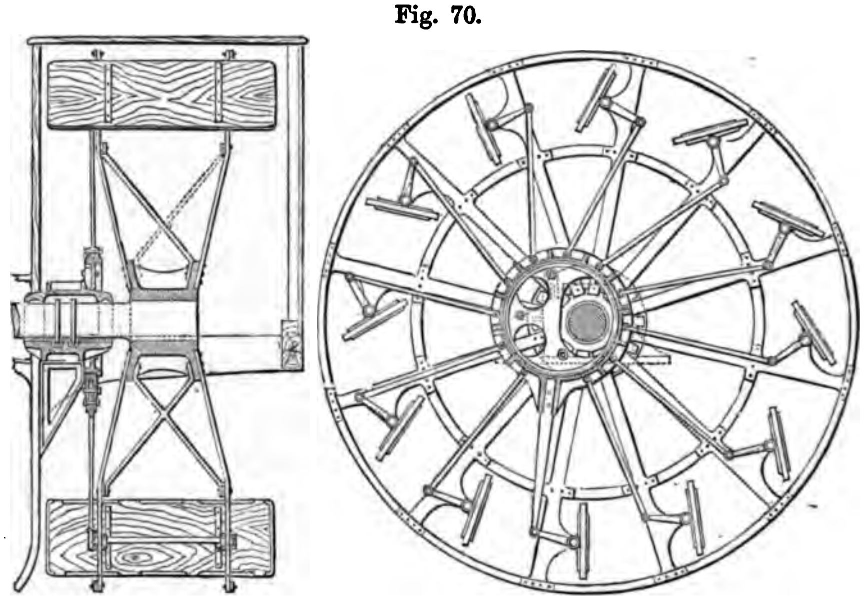
the consideration of the marine engineer. The speed and displacement of the hull must be noticed, the area of the floats immersed, and other matters already alluded to; and, even with all these, rarely does theory and practice agree on all points.

Up to the present, two kinds of paddle wheels are in general practice, radial and feathering—these terms referring especially to the floats; the radial, which has fixed floats, while for the feathering kind the floats oscillate.

To render this matter obvious to the uninitiated, and of practical utility to the conversant, the illustration, Fig. 70, is introduced, being an example in connection with plate 30.

The centre piece is seen both complete and in section. It will be noticed that each arm is recessed in the flanges, and the rigidity of the connection is ensured by keys, bolts, and nuts. Each arm radiates from the centre of the paddle shaft, and is prolonged, at the same angle, to the outer ring. The connection of the arms with the rings is attained by bolts and nuts, suitable projections being formed on each arm. The float-shafts are supported within projections forged with each arm, between the rings. The floats are sustained on the shafts by clamps, bolts, and nuts, and levers keyed on the inner extremities of the shafts are connected with the radius-rods. In the side elevation each float is represented at different angles, and the lowest float is in a vertical position. This float is connected by a lever and rod to an eccentric band, and the rod *fixed* in the band is termed the driving rod. The rod in question is inserted in a provision on the band—seen also in the sectional elevation, and bolts and nuts complete the connection. The band is in two parts, connected in the ordinary manner; and ribs are cast between each radius rod to ensure strength. To reduce the friction, and prevent rust, the band and eccentric are lined and

Fig. 70.



FEATHERING PADDLE WHEEL.

surfaced with gun-metal rings. Each radius rod, lever eye, float shaft, clamp eye, and their respective pins, are similarly fitted. The eccentric is of course stationary, and secured by bolts and nuts to a provision formed with the plummer block. This latter detail being beyond the eccentric, is depicted by dotted lines in the side elevation, but in section in the other view. The paddle shaft has formed on it collars, to prevent lateral disturbance, and straining the entablature. This is a great advantage over the plain surface bearings, and the credit of the improvement is doubtless due to the firm in question—Messrs. James Watt and Co. The cap has brass only at the extremities, as the friction is mostly —if not entirely—on the lower half of the bearing.

The correct arrangement of the arm stays for paddle wheels is no mean attainment, and in the present example attention has been bestowed on this matter. These stays are seen in the sectional elevation; those connected to the lower arms are nearest the driving rods; the upper, or remainder of the arms, being fitted with single stays at alternate angles—one being shown complete, and the other in dotted lines. The centre piece is strengthened by ribs joining, or between, the flanges, and thus great strength is effected. The arms are stayed also at the projections supporting the float shafts by cross bolts and stays, one of which is seen in front of the float in sectional elevation: these stays are situated directly behind each float shaft.

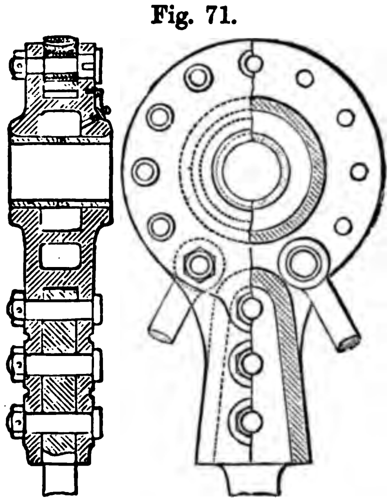
As dimensions are always worthy of attention, and—indeed, with correct proportions of proved effect—are invaluable as a reference, the following are inserted, being in connection with the arrangement under notice:—

	Ft.	In.
Diameter of Polygon	19	9
Number of Floats, 12.		
Length of Float	10	0
Width of ditto	3	3
Thickness of ditto	0	3½
Number of Floats immersed at the same time, 3.		
Depth of immersion to centre of Float	3	7
Outside diameter of outside Ring	24	0
Ditto ditto of inside Ring	15	9
Space between Rings	3	9
Width of outer and inner Rings	0	4½
Thickness of ditto	0	0⅞
Width of Arms	0	6
Thickness of ditto	0	0⅞
Length of Connecting Portions	1	3
Length of projections from side of arm to centre of Float Shaft	1	9
Diameter of Float Shaft	0	3
Length of ditto	6	0
Length of Clamps	2	6
Width of Clamps	0	4
Distance from centre of eye to edge of Float	0	7
Length of Float Levers	1	9
Diameter of Pins	0	2
Diameter of Radius Rods	0	2½
Length of Radius Rod	6	9
Distance between centre of Driving Lever to centre of Eccentric	9	7
Diameter of pitch line of suspension circle in Eccentric	5	8
Throw of Eccentric	1	2
Rise of centre of Eccentric	0	1½
Diameter of Eccentric band-surface	5	0
Diameter of band Bolts	0	2
Width of Band	0	7
Width of Eccentric	0	9
Width of Driving Rod at eye	0	3
Thickness of ditto at ditto	0	1½
Diameter of Stay at Driving Arm	0	1⅜
Ditto of single Stays	0	1⅜
Diameter of cross Stays	0	1⅜
Transverse distance between centres of Arms	5	3
Length of Float Shaft Boss	0	5
Transverse distance between centre of Eccentric and centre of Floats	3	1
Diameter of Centre Piece	5	0
Extreme width across Flanges	3	6
Thickness of Flanges	0	2¾

	Ft.	In.
Thickness of Boss	0	5½
Length of Boss	3	0
Diameter of Paddle Shaft at Boss	1	9
Length of bearing of shaft in Plummer Block	3	6
Diameter of Shaft	1	6
Space between Collars	0	6
Width of Collars	0	2
Diameter of Collars	1	11½
Thickness of lower brass	0	1½
Ditto of Cap brasses	0	0¾
Width of ditto	0	5
Height of Block	1	7
Length of sole Plate	6	7
Width of ditto	1	9

The eccentric motion for feathering the floats is not, in all cases, as that last alluded to. It is often preferred by some engineers, among whom may be mentioned Messrs. Ravenhill and Hodgson, to permit the eccentric to revolve on a separate shaft, and thus dispense with the band, also reduce the diameter of the eccentric. Obviously, with this mode of connecting the radius rods, the entire arrangement must be situated outside the paddle-centre rather than inside, as in the example illustrated by Fig. 70—page 240.

Fig. 71.

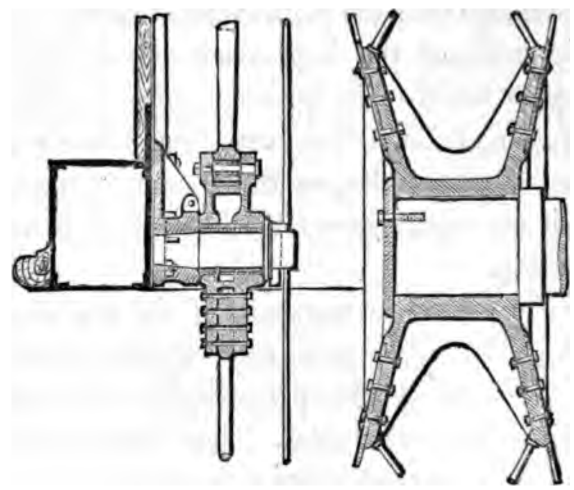


PADDLE WHEEL OUTSIDE ECCENTRIC.

The illustration, Fig. 71, is an eccentric of the class under notice. The shaft on which it is supported is secured by a bracket to the beam of the paddle box. The required oscillation of the floats is attained by securing the shaft at a certain distance from the centre of the paddle shaft, and the exact point is determined precisely as that for the eccentric in the previous example.

The sectional elevation represented by Fig. 72, shows the application and position of

Fig. 72.



APPLICATION OF OUTSIDE ECCENTRIC.

an eccentric similar to Fig. 71, in relation to the extremity of the paddle shaft. The remaining particulars of the details being duplicates of those alluded to in connection with Fig. 70—page 240—further description is unnecessary.

The mode of securing the fixed floats to the arms is by hook-end bolts, and ordinary nuts; the curved portion of each bolt clasps the arms, and the other extremity passes through the clamps and float.

It is not essential with all examples of paddle wheels that the arms should be inserted or secured to the centre piece. In

some instances an intermediate or alternate connection only is requisite; and short arms, from the inner to the outer ring, complete the entire frame. This latter arrangement is often used with fixed floats, and also in some cases with the feathering floats. With those, in some instances, the outer ring is dispensed with, and the projections on the arms, alone, support the floats and requisite details. This, however, is not to be universally recommended, although material and labour are reduced by its adoption.

With reference to the centre pieces, they are generally as these shown by Figs. 70 and 72—pages 240 and 242. The mode of securing them on the shaft is by keys, and, in some instances, lateral displacement is guarded against by a plate secured by studs at the end of the shaft—also shown by Fig. 72. The means for turning the engines when "cold," or not under steam, is by securing a toothed ring to the outer or inner ring —but mostly the former; and an ordinary pinion gearing with the same, imparts the required motion by hand power—the handle being beyond the deck side of the paddle-box.

It is of course known to all engineers, that design, and even arrangement, is subject to the power to be exerted; and the mode of connection, with a small example, will admit of various modes not applicable in a large type. It is for this reason, mostly, that such trivial alterations and variations are in being, not being therefore as much thoughts of caprice as ideas of circumstance.

CHAPTER VI.

DETAILS OF ENGINES FOR SCREW PROPULSION.

THE "screw engine," as it is usually termed, has many advocates of various opinions, inasmuch that rarely, even with the details, do two makers agree as to the design. Of course, as to the principle, there can be but one aim in view; the difference, therefore, pertains to the mechanical question. This much, however, must be acknowledged, that if by adverse opinions, many modes—each producing the same result—are originated, no little credit is due to those engaged thereon. When cogitating on a new idea for a given purpose, the mind should not be swayed in one direction; rather should the defects be remembered than set aside, by which means a truthful conclusion must be arrived at. Two facts should be ever before the designer—"simplicity of construction," and "access for repair." Now it may be urged that the latter consideration is an admission of a defect, to be retained, rather than obviated. It can be said in answer, however, that natural laws have not, to the present, been fathomed deeply enough to ignore repair, and that more wisdom is displayed by the general who deliberately prepares for the reception of the enemy, than he who relies on sudden action. Of course, confidence in the matter in hand must not be omitted, for without self-credit no inventor or schemer can proceed. A sure means of success is to acknowledge the faults committed, and obviate them in the next example, rather than be led by conceit in the accomplishment of one object.

In pages 38 to 75, the variations in the arrangements of the screw engines have had attention: the present chapter is in connection with the same, in direct allusion to the details of the previous examples.

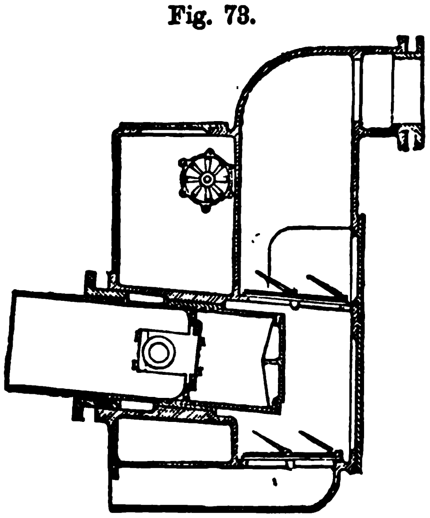
The condenser of a horizontal engine determines the lateral disposition of the cylinders, obviously so, on remembering that when the pumps are worked by the steam piston, the centres of the former must be within the circle of the cylinder's area. It is with a just recognition of this fact that the condensers will have prior notice in this chapter.

INJECTION CONDENSERS, WITH TRUNK AIR PUMPS.

The type of condenser first to be noticed is fitted with the trunk air pump. Of this class, two firms only—one in England and the other in Scotland—at present adopt the same. The illustration, Fig. 73—page 245—is an elevation of the condensers of the engine illustrated by Fig. 4—page 44. The plunger is a hollow tube, with a closed end, forming a separate portion, secured by bolts and nuts. The connecting-pin of the con-

necting rod is secured, about two-thirds of the length of the plunger, by bolts and nuts, to flanges cast with the barrel, so that when the plunger is at half-stroke—as shown—the centre of the pin is at the centre of the guiding and packing portions forming the stuffing box. The condensing diameter is at the

Fig. 73.



MESSRS. RENNIE'S TRUNK AIR PUMP INJECTION CONDENSER.

front end of the structure. The injection valve—seen near the top—is a perforated disc, acting on a seating of similar design. The exhaust steam enters the condenser opposite the injection valve, and thus an immediate contact is the result. The door above the valve permits inspection and access for repairs. The pump being single acting, the valves are at the extreme end of the barrel, over and under the same. These valves are almost square in plan, and the seatings perforated like the honeycomb, rather than square or round passages. The guards are flat perforated plates, at the requisite angle, in proportion to the lift requisite for the valves. The doors for the inspection, renewal, &c., are at the back of the discharge chamber. This latter compartment is over, and partly around, the barrel. The discharge opening is above the roof of the condenser, rather than level with it, as shown in Fig. 44. This is one of the latest examples by the Messrs. Rennie, fitted with the engines in H.M.S. "Reindeer" —since the International Exhibition of 1862, where it was shown. The condenser with the discharge passage "below the roof" was fitted in H.M.S. "Ranger," in 1860.

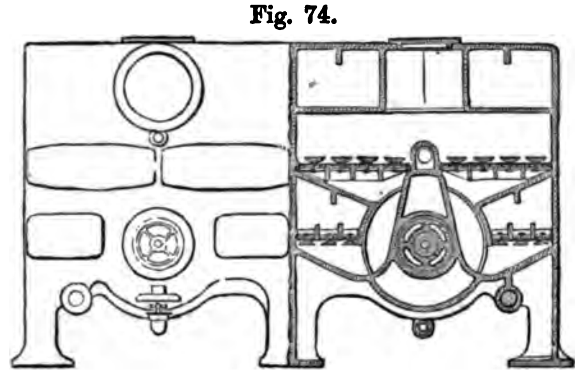
The snifting valve in the section under notice is at the front of the condenser, while in other examples the back of the same is considered the preferable position; certainly the latter position is the more accessible.

The following dimensions are those from which the illustration, Fig. 73, was produced; being one of the duplications for a pair of engines of 200 horse power nominal collectively.

	Ft.	In.
Diameter of Pumps, 2	1	6
Length of Stroke	2	0
Diameter of Connecting Rod . . .	0	3
Sizes of Valves (7 × 12 in.)		
Number of Suction ditto, 6.		
Number of Discharge ditto, 6.		
Diameter of Injection Pipes, 2 . . .	0	3¼
Diameter of Discharge Pipes, 2 . . .	1	0

The next example of trunk air pumps and injection condensers to be described is that

Fig. 74.



MESSRS. NAPIER'S TRUNK AIR PUMP INJECTION CONDENSER.

represented by Fig. 74. This is a transverse

section and back and end elevation of the condenser and pumps alluded to by Fig. 5—page 47. The plunger, although connected to the crank pin—as in the example—receives its motion from the steam piston. The plunger is closed at the inner end, while a single action only is attained. The bottom of the condenser is level with the centre of the air pump, and the suction valves are inverted. The discharge valves are above the pump, also the chamber. The condenser extends from the centre of the pump to the roof, the passages for the suction and discharge being at the back of the arrangement. The section illustrated is taken beyond the plunger, to show the means of guiding the same, at the back end, by a prolongation of a smaller diameter than the pump. This extension works through a stuffing box and gland—seen in the complete view. The gain effected is of a two-fold character, not only guiding, but also retaining the horizontal line of the trunk, when the same is at the complete out-stroke. Exclusive of this, the connecting rod can be adjusted in its bearing, from the back end of the condenser, while the engine is in motion, if desirable. This is attained by an adjusting rod passing through the tube and eye rod. The adjusting rod is screwed in its central bearing, and thus any motion imparted to it causes an advancing or receding action. This is illustrated longitudinally in Plate 16, to which reference is made in the description of the plates.

The snifting valves are at the back of the structure, directly under the pumps, separate pipes connecting therefrom with the bottom of the condenser at the front end. The exhaust steam enters the condensing chamber at the front end, below the roof—fore and aft of the pumps. The injection water meets the steam from the roof, the distribution of the fluid being effected through a spray plate suspended therefrom. The discharge from the pump is through the passage above the valves, seen in the complete view.

Now with reference to the efficiency of the arrangement under notice, doubtless the principle aim is, to engross all the requirements of a condenser within the least possible space, and adopt the trunk obligatorily, as a means of transmitting motion to the crank pin.

Messrs. R. Napier and Sons have fitted many vessels with these condensers, and amongst their late productions that fitted in H.M.A.P.S.S. "Hector," of 900 horse power nominal collectively, as illustrated by Fig. 74—page 245—the principal dimensions of which are as follows:—

	Ft.	In.
Diameter of Trunks, 2	3	7
Diameter of Guiding or Adjusting Tube .	1	6¾
Length of Stroke	4	0
Diameter of Connecting Rod . . .	0	8¼
Diameter of Suction Valves . . .	0	6½
Number of ditto, 42.		
Diameter of Discharge Valves . . .	0	9⅛
Number of ditto ditto, 24.		
Diameter of Injection Pipes, 2 . . .	0	7½
Diameter of Discharge Pipes, 2 . .	2	1

INJECTION CONDENSER FOR DIRECT ACTING ENGINES.

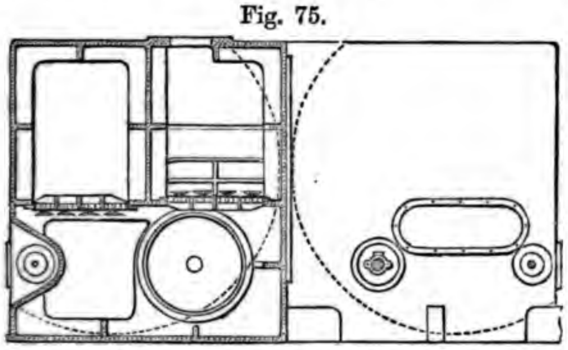
The type of condenser now under notice is the ordinary kind, fitted with double acting air pumps; and in the place of a plunger a piston is used, and a rod connects the same to that in the cylinder.

With this arrangement of direct acting

engine every facility is available to arrange the condenser pumps and valves as compact as practicable, and this advantage is well represented by the following examples.

Messrs. J. Penn and Son arrange the valves and pumps as illustrated by Fig. 75. The con-

Fig. 75.



MESSRS. PENN'S INJECTION CONDENSER FOR TRUNK ENGINES.

densing and discharge chambers are above the periphery of the pump. The suction valves are inverted at the side of the barrel, and those for the discharge are directly above, with upward action. The barrel is a tube cast midway of the length; each set of valves is arranged to drain and discharge simultaneously —or when the front end set is draining the condenser, those at the opposite extremity are permitting the final discharge. The arrangement under notice is in duplicate portions, thus rendering each condenser, pump, and set of valves common only to the opposite engine. There are two practical advantages resulting from this; first, a saving of patterns, and secondly, portability for erection. Further than this, another gain is effected, *i.e.*, in the event of either engine being disabled, the one in repair is not encumbered with both condensers, sets of valves, &c.

A longitudinal section of the arrangement will show the valves arranged the entire length of the condenser, in sets of four and two rows, on each side of the central rib, supporting the barrel; and this in principle relates also to the succeeding examples.

The exhaust steam enters each condenser at the top, or on the roof of the condensing chamber. The injection water is introduced at the side—fore and aft—directly below the roof, therefore an instantaneous effect is ensured. The discharge opening is nearly level with the bottom of the chamber over the pump. The doors for access to the suction valves are at the back and front of the chamber below the same; those for the discharge valves are on the roof, near the central side or connection of the structure.

From the illustration and description it is obvious that compactness of disposition, for the detail, is, combined with correct localization, characteristics common with the firm in question.

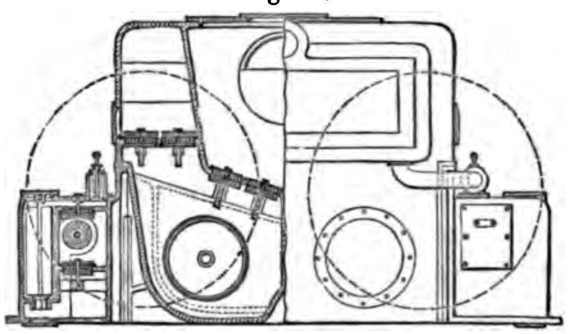
For engines of 1,350 horse power nominal collectively, the diameters of the pumps, valves, &c., are nearly as follows:—

	Ft.	In.
Diameter of Pumps, 2	2	10
Length of Stroke	4	4
Diameter of Pump Rod	0	4¾
Diameter of Suction Valves . . .	0	8½
Number of ditto ditto, 48.		
Diameter of Discharge ditto . . .	0	8½
Number of ditto ditto, 48.		
Diameter of Injection Pipes, 2 . . .	0	9
Diameter of Discharge Pipes, 2 . .	2	0

Next on the list of originators of well-arranged air pumps and condensers is the well-known firm of Messrs. Humphrys and Tennant. They, unlike Messrs. Penn, prefer one condensing chamber common to both cylinders, as shown by Fig. 76—page 248. The discharge chambers are on each side of

the condenser, and both are sufficiently high above the pumps to enable a free passage for the water. At the back end of the condenser the discharge chambers are connected by passages, and thus one discharge pipe only

Fig. 76.



MESSRS. HUMPHRYS' INJECTION CONDENSER FOR DIRECT ACTING ENGINES.

is requisite. The suction valves are inverted in the bottom of the condenser at an angle; and the discharge valves, with a reverse action, are secured in the bottom of the discharge chambers.

The valves are the flat kind, almost square in plan—the seating being perforated in the ordinary manner. The guards are twin-angular, and thus the valve rises and falls on each side of its connection.

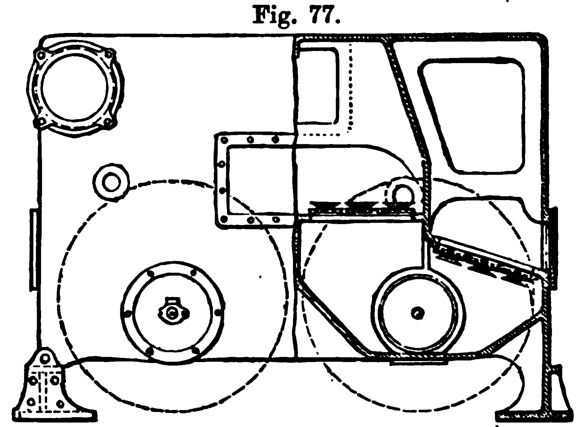
The exhaust steam enters the condenser at the front end, below the roof, and the injection water directly opposite. The doors for access to the valves are at the sides and top of the respective chambers.

The following dimensions are the general practice of the firm in question for a pair of engines of 400 horse power nominal collectively:—

	Ft.	In.
Diameter of Pumps, 2	1	10
Length of Stroke	3	0
Diameter of Pump Rod	0	$2\frac{3}{4}$
Size of Valve (9 × 7 inches.)		
Number of Suction Valves, 48.		
Number of Discharge Valves, 48.		
Diameter of Injection Pipe	0	8
Diameter of Discharge Pipe	2	0

The arrangement next under comment is that by Messrs. J. and W. Dudgeon, represented by Fig. 77. The central compartment is the

Fig. 77.



MESSRS. DUDGEON'S INJECTION CONDENSER FOR DIRECT ACTING ENGINES.

discharge chamber—common to both pumps—and the condensers are on each side of the same. The suction valves, of the disc kind, are inverted at an angle, as in the previous example; also the disposition of the discharge valves are similar. The exhaust steam enters the front of the condensers below the roof, and the injection water flows in at the sides, to render a vacuum certain.

This arrangement may be said to be a combination of those represented by Figs. 75—page 247—and 76—in relation to the positions of the respective chambers, pumps, &c.

The doors, for internal access to the suction and discharge valves, are at the front, back, and sides of the chambers. The discharge opening is at the back end and directly below the roof.

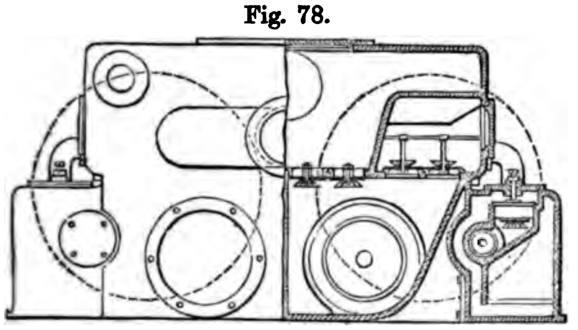
The example under notice has been fitted by Messrs. Dudgeon in the mail steamer "Mary Augusta." The engines are 300 nominal

horse power collectively, driving twin-screws, and the proportions of the details under notice are thus :—

	Ft.	In.
Diameter of Pumps, 2	1	1
Length of Stroke	1	10
Diameter of Pump Rod	0	2½
Diameter of Suction Valves	0	6½
Number of ditto ditto, 36.		
Diameter of Discharge Valves	0	6½
Number of ditto ditto, 36.		
Diameter of Injection Pipes, 2	0	4
Diameter of Discharge Pipes, 2	1	3½

If not in design, certainly in arrangement, the illustration depicted by Fig. 78 forms a

Fig. 78.



INJECTION CONDENSER FOR DIRECT ACTING ENGINES.

striking similarity to that of Fig. 76. The situation of the condenser is the same, also the discharge chamber, but the injection water is admitted at the sides—at the back end, rather than a central position. Both discharge chambers are connected at the back ends by a passage, and the discharge pipe is connected to the central opening.

The present example is compiled from a pair of engines of 60 nominal horse power collectively, to which the following dimensions pertain :—

	Ft.	In.
Diameter of Pumps, 2	0	8¾
Length of Stroke	1	2
Diameter of Pump Rod	0	1
Diameter of Suction Valves	0	4
Number of ditto ditto, 16.		
Diameter of Discharge Valves	0	4
Number of Discharge Valves, 16.		
Diameter of Injection Pipes, 2	0	2¾
Diameter of Discharge Pipe	0	6

INJECTION CONDENSERS FOR RETURN ACTION ENGINES.

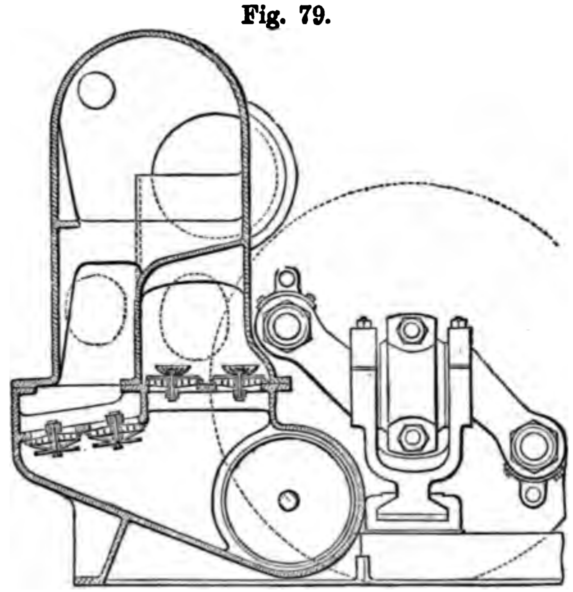
The type of engines, in connection with the condensers now to be noticed, interfere with the arrangement of the pumps and valves to a considerable extent. The steam piston rods have to be prolonged at the sides of the condenser, and the guides must be in close proximity. Obviously then, the authorities are to be commended for their different productions, each accomplishing the same result by dissension of opinion. As a proof of this, it must be remembered that each maker deals with the same evils common with the type of engine; therefore, no chance for escape or obviation is possible. Now with direct acting and trunk engines, all makers of these examples can freely employ the most compact arrangement of condenser, &c., and thus the cylinders can be situated at the least possible space apart.

It is current amongst all who have studied this matter, that two compartments are essential to produce condensation, such as condensing and discharge chambers, and that the former should be above the air pumps, &c. The engineer has, therefore, to consider which chamber shall be centrally placed in the arrangement, or rather which shall be fore or aft of the connecting and piston rods. In conjunction with this, attention has to be given to the transverse space required for the valves, and the fact also that the air pumps must be worked by the steam piston is often the cause of much trouble to attain a conclusion.

The matter has resolved itself into three questions, as preliminaries. First, shall the condenser be between the *inner* piston rods of each engine, or beyond those outside the guides?—the discharge chamber being, of course, similarly affected; secondly, shall both chambers be centrally placed? and thirdly, shall the entire arrangement be outside or fore and aft of the rods?

No firm has devised more varied examples of the type of condenser under notice than Messrs. Maudslay, Sons, and Field, and their latest and best production is illustrated by

Fig. 79.



MESSRS. MAUDSLAY'S INJECTION CONDENSER FOR RETURN ACTION ENGINES.

Fig. 79. The air pump's centre is within the periphery of the cylinder, and the entire arrangement of chambers, valves, and pumps, is outside the fore and aft piston rods. The condenser is of peculiar form—common to the requirements determined—although the discharge chamber may almost be deemed an intrusion in the former compartment. The suction valves are inverted above the pump, and those for the discharge are correctly arranged in the bottom of their respective chamber. The exhaust steam enters the condenser directly above the partition or roof of the discharge chamber, and the injection water is admitted near the roof of the condenser. Suitable doors are secured at the end of the structure to inspect or renew the valves—the discharge water, of course, passes out at the back end between the roof and base of the chamber.

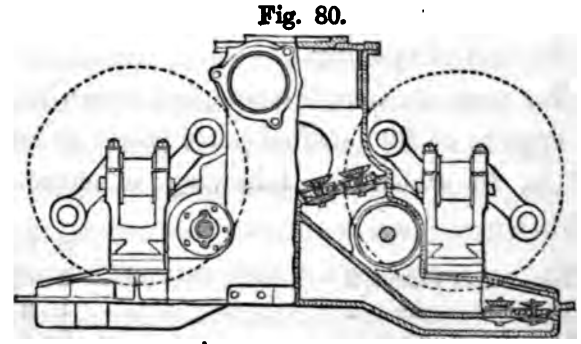
This arrangement of the details is as compact an arrangement as almost possible, admitting the late improvements, and that practice in the hull has not belied the theory founded in the drawing office.

For a pair of engines, 900 horse power nominal collectively, the proportions by the firm are thus:—

	Ft.	In.
Diameter of Pumps, 2	2	2½
Length of Stroke	4	0
Diameter of Pump Rod	0	3¾
Diameter of Suction Valves	0	10¾
Number of ditto ditto, 68.		
Diameter of Discharge Valves	2	10¾
Number of ditto ditto, 72.		
Diameter of Injection Pipes, 2	0	7
Diameter of Discharge Pipes, 2	1	1¾

For engines of moderate power, or about 200 horse power nominal collectively, the

Fig. 80.



MESSRS. MAUDSLAY'S INJECTION CONDENSER FOR RETURN ACTION ENGINES.

Messrs. Maudslay adopt the arrangement de-

picted by Fig. 80. In this example the suction valves are below the air pump, and those for the discharge above. The condensing compartment is at the front of the arrangement, and extends for the entire length under the air pump. The discharge chamber is centrally situated between and above the pumps. The exhaust steam is admitted at the front end, and the injection of water directly above. The discharge from the chamber is effected at the back end directly above the valves.

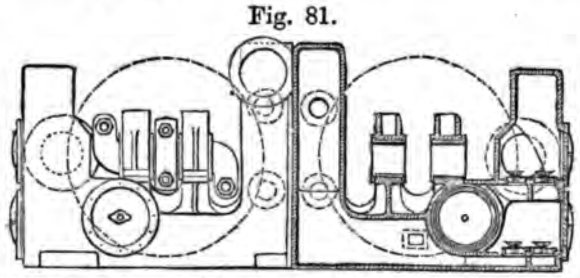
It will be noticed that due attention has been directed, by the designers, to the accessibility to the valves—the doors, for that purpose, being above each set rather than at the side of each compartment. It is also apparent that, with this and the previous example, the steam piston and air pump rods are directly above each other, thus retaining a similar position for the pumps in relation to the cylinders, which is one of the important features in the subject under notice.

For a pair of engines of 200 nominal horse power collectively, the following proportions are the practice by the firm in question :—

	Ft.	In.
Diameter of Pumps, 2	1	$1\frac{1}{4}$
Length of Stroke	2	0
Diameter of Pump Rod	0	3
Diameter of Suction Valves . . .	0	7
Number of ditto ditto, 20.		
Diameter of Discharge Valves . . .	0	7
Number of ditto ditto, 24.		
Diameter of Injection Pipes, 2 . . .	0	$4\frac{1}{2}$
Diameter of Discharge Pipe . . .	1	2

Messrs. Ravenhill and Hodgson are now to be alluded to, in connection with the present subject. This firm for some time adopted the arrangement represented by Fig. 81. In this case the condensers are separate, centrally located, and extend under the guides for the piston rods. Almost in a vertical line below the outside piston rod, the air pump is situated, deriving its motion from the steam piston. The suction valves are level with the bottom of the pump at the side, and those for the

Fig. 81.



MESSRS. RAVENHILL'S INJECTION CONDENSER FOR RETURN ACTION ENGINES.

discharge directly above. The discharge chambers are therefore outside the piston rods, and thus both compartments form—by their positions—a channel for the main motion of the engine.

The exhaust steam enters the chamber at the front end near the roof, and the injection water is admitted directly below at the opposite end. The doors for the various purposes are at the sides of the respective chambers, and thus the entire arrangement may be termed compact and efficient.

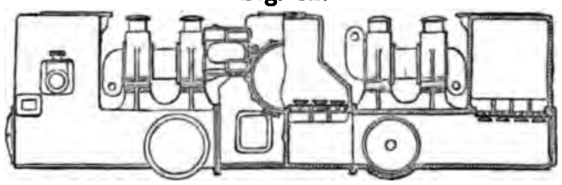
The proportions adopted by the firm alluded to for a pair of engines of 250 nominal horse power collectively, are as follows:—

	Ft.	In.
Diameter of Pumps, 2	1	$3\frac{3}{8}$
Length of Stroke	2	6
Diameter of Pump Rod	0	3
Diameter of Suction Valves . . .	0	8
Number of ditto, 24.		
Diameter of Discharge Valves . . .	0	8
Number of ditto, 24.		
Diameter of Injection Pipes, 2 . . .	0	3
Diameter of Discharge Pipes, 2 . . .	1	$1\frac{1}{2}$

The correct position for the condenser is—as often before stated—above the air pump,

and the firm in question have duly recognized this by their arrangement depicted by

Fig. 82.



MESSRS. RAVENHILL'S INJECTION CONDENSERS FOR RETURN ACTION ENGINES.

Fig. 82. Here it will be noticed the suction and discharge valves are above the pump, each set level with each other, and in reverse localities to those in the previous example. The chambers also are in contrary positions in relation to the piston rods and guides.

The condensers are fore and aft of the piston rods, in separate compartments common to each cylinder. The discharge chamber is centrally placed, or between the pumps over the same, and thus a single compartment is adopted in the place of duplications.

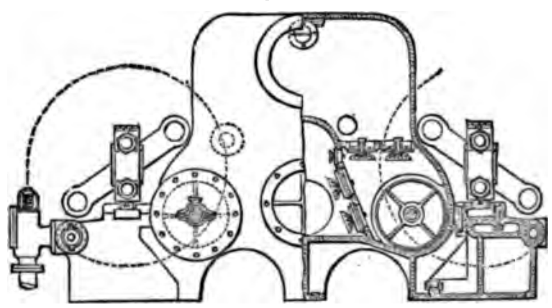
The exhaust steam enters each condenser at the roof, near the front end, and the injection water valve is at the back end. The position of each air pump in relation to the piston rods is almost central, rather than as the prior examples, but the great distance between the valves causes a great "wash" on each side of the pump. The discharge pipe is secured at the back end directly above the valves. The means of inspection is provided for by doors, situated on the roof of the discharge chamber, and at the sides of each compartment below the condensers.

Messrs. Ravenhill and Hodgson have fitted many ships of large tonnage with this arrangement. Amongst them for the present purpose is selected that represented by Fig. 82, the dimensions being in relation to a pair of engines, 900 nominal horse power collectively:—

	Ft.	In.
Diameter of Pumps, 2	2	1¼
Length of Stroke	4	0
Diameter of Pump Rod	0	3¾
Diameter of Suction Valves	0	9
Number of ditto ditto, 96.		
Diameter of Discharge Valves	0	9
Number of ditto ditto, 96.		
Diameter of Injection Pipes, 2	0	8
Diameter of Discharge Pipe	2	5½

Attention is most directed to the arrangement illustrated by Fig. 83. This is originated by ourselves, the principal aim being to

Fig. 83.



BURGH'S INJECTION CONDENSER FOR RETURN ACTION ENGINES.

adopt the advantages common with the condensers for direct acting engines and apply them to the return acting types.

It will be noticed that the condenser is a single compartment above the air pumps. The discharge chamber is under the condenser, between the pumps, and by this arrangement space is economized. The guide channels are beyond the structure—being a reverse position to that adopted by the Messrs. Maudslay, shown by Fig. 79—page 250. The exhaust steam is admitted for

condensation below the roof of the chamber, at the front end, opposite to which is the injection valve. The position of the suction valves are similar to that in Fig. 75—page 247—but the discharge valves are almost vertically secured in the chamber — being unlike any of the previous examples alluded to. To ensure the air escaping on being discharged at each stroke of the piston, "air valves" are secured above the main valves. The discharge water passes out at the back end of the chamber. The doors for access to the suction valves are those of the air pump barrel, and those for the discharge valves are at the back and front ends of the respective chambers. In some instances the suction valves have separate doors at the sides directly above the pumps.

The exact position for the valves as shown, need not be retained, as with larger examples much more space is available. For instance, the base of the condenser may be inclined as shown by Fig. 76—page 248—and thus nearer to the pump—always remembering that space must be retained for securing and renewal of the detail. The discharge valves also can be more angularly or vertically secured, provided the sides of the chamber are thus formed.

The author's proportions for a pair of engines 200 nominal horse power collectively, are thus :—

	Ft.	In.
Diameter of Pumps, 2	1	2¾
Length of Stroke	2	0
Diameter of Piston Rod	0	2⅜
Diameter of Suction Valves	0	6½
Number of ditto, 16.		
Diameter of Discharge Valves	0	6¾
Number of ditto, 16.		
Diameter of Injection Pipe	0	3¾
Diameter of Discharge Pipe	1	3½

SURFACE CONDENSERS.

As far back as the year 1832 Mr. Hall—since dead—proved in the most practical manner that surface condensation was not only efficient but also economical. Engineers and shippers were, however, slow to believe in the matter, and thus until 1862—a lapse of thirty years—the process died an easy death. Certainly here and there a spark of its existence was to be found, but in the main its termination was effected.

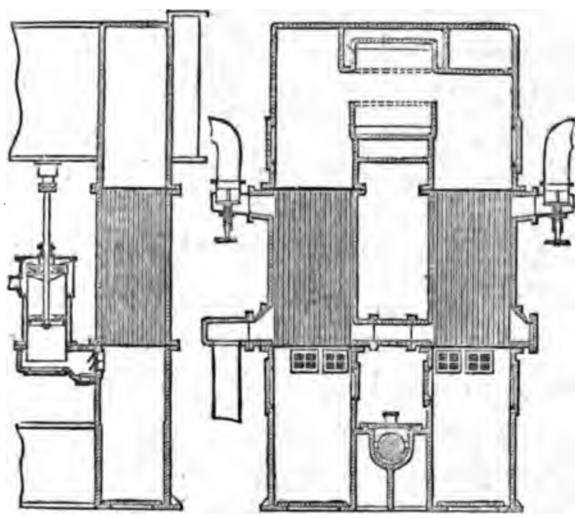
At the International Exhibition of the period last alluded to, makers came forward with examples of surface condensers, and being ready to stand by their productions—the system revived, or rather the authorities on the subject stirred themselves. The result is that the surface condenser is *now* an universal appendage in connection with the marine engine of the present day.

The best mode of adopting the system under notice is, of course, an interesting question, and its solution cannot be better expressed than by the paper on "surface condensation in marine engines," delivered by the late Mr. E. Humphrys, then principal partner in the firm of Messrs. Humphrys and Tennant, in the year 1862, to the members of the "Institution of Mechanical Engineers." After paying a graceful tribute of recognition to Mr. Hall, as the founder of the function he was about to treat of, Mr. Humphrys proceeded to state, in allusion to the arrangement shown by Fig. 84—page 254—

That it showed very nearly the arrangement of the condensers of the "Mooltan;" and it also showed correctly the condensers then making by the writer for the Peninsular and

Oriental Co.'s ships "Mysore" and "Rangoon," of 400 nominal horse power. The area of surface in the condensers and in the boilers, said Mr. Humphrys, of all the three ships, is

Fig. 84.



MESSRS. HUMPHRYS' SURFACE CONDENSER FOR VERTICAL ENGINES.

almost identical: the boilers contain 4800 square feet of heating surface in each ship, and the condensers of the "Mysore" and "Rangoon" contain 4712 square feet of condensing surface, and those of the "Mooltan" 4200 square feet. The indicated power of the "Mooltan" when tried officially was 1734 horse power; hence the area of the condensing surface per indicated horse power is rather less than $2\frac{1}{2}$ square feet.

For convenience of manufacture and arrangement of these engines, the condenser of each is divided into two parts, each part being exhausted by its own air pump, so that each pair of engines is provided with four air pumps and four condensers. The air pump is 18 inches in diameter, with a stroke of 3 feet. These dimensions being used by the writer with injection condensers in engines of the same nominal power, he believes they are larger than necessary for surface condensers of engines in good condition, with condensing water at the average temperature of the sea in this climate; but as these engines are to be employed in the Indian seas, it was considered expedient to provide large air pumps and large pumps for circulating the condensing water, so as to allow for almost any quantity of condensing water being driven through the condensers that may be found necessary in an Indian climate. The air pumps discharge their water direct into the boilers through the pipe, according to "Hall's" plan, so that no feed pumps are necessary. The air which leaks into the engines is allowed to escape by an open stand pipe connected to the highest point of the feed pipe, and carried up inside the mast, which is of iron, to a greater height than is due to the pressure of steam in the boilers. A valve regulated by a float was originally fitted to the "Mooltan," for allowing the escape of the air; but it was found to require some little attention, and hence the stand-pipe was substituted, which answers perfectly without any attention.

Each condenser contains 1178 seamless drawn pure copper tubes, $\frac{5}{8}$ inch outside diameter and No. 18 wire-gauge or ·05 inch thick, 5 feet 10 inches long, weighing 28 ozs. each tube, and fixed at 1 inch pitch centre to centre. The tube plates of the "Mooltan" are of cast gun-metal $\frac{3}{4}$ inch thick; but those of the "Mysore" and "Rangoon" are of rolled copper, finished $\frac{3}{4}$ inch thick. These are first set as flat as possible, and the tube holes marked out upon them. The holes are then drilled under a common drilling machine with a drill of two diameters, having a guard upon it to fix the depth to which the

larger diameter shall penetrate the plate. One machine worked by an ordinary driller drilled the 1178 holes in the tube plate in 70 hours. The tapping of the holes is then proceeded with, and is effected with a tap, having a parallel end to guide it, which fits the smaller diameter of the tube holes. One man of ordinary skill tapped the 1178 holes in the plate in 70 hours. After having been drilled and tapped, the tube plate is again set perfectly flat on a surface plate, and then both sides are faced off in a lathe or planing machine.

The screwed glands, for securing the packing at the ends of the tubes, are made from Muntz metal solid rolled tubes, which are obtained in lengths of about 5 feet, rolled to gauge both inside and outside; the inside diameter is exactly that of the outside of the copper tubes, namely $\frac{5}{8}$ inch, and the outside diameter is such that when screwed it will exactly fit the tapped holes in the tube plates. It is screwed on the outside as it comes from the maker in a common screwing machine, and is then cut by a circular saw into half-inch lengths to form the glands. The saw marks are taken off the ends by a facing cutter revolving on a lathe, and the same operation clears out the inside of the hole. The notch for the screwdriver is cut by passing a number of the glands, when screwed into a plate, under a revolving circular saw of the required thickness. The packing is composed of linen tape; a piece of this tape 12 inches long, $\frac{7}{16}$ inch wide, is wound round a mandril, the ends and edges being slightly stitched, in which state it is readily put into the tapped holes of the tube plate, and when screwed down by the gland forms a very perfect and lasting joint. The thickness of the tape is such that 1300 of these packings weigh about 2 lbs.

The exhaust steam from the engines passes down through the interior of the condenser tubes, and the sea water for keeping the tubes cool is driven up through the spaces between the tubes. The sea water is admitted through an inlet pipe fitted with a slide valve at the bottom of the ship, and enters at the bottom by the pipe. It then circulates round the outsides of the tubes, and makes its exit through the regulating valves at the top of the condensers, at about the load water line of the vessel. The disc valves shown answer the purpose of regulating the flow of sea water equally through the two divisions of the condenser, and also of shutting out the water from above, when the outsides of the condenser tubes have to be examined. The flow of water is produced by one of Appold's centrifugal pumps, the diameter of the revolving disc being 36 inches. It is driven by a pair of wood and iron spur wheels, the proportions of which are 1 to $3\frac{1}{2}$, so that at the ordinary speed of the engines of the "Mooltan," namely 56 revolutions, the pump makes 194 revolutions per minute. Two of these pumps are provided, the second being driven by an auxiliary engine, to be used in case of the failure of the other.

Making allusion to the means of making a perfect joint for the tubes, Mr. Humphrys observed:—Before determining on adopting exactly "Hall's" mode of manufacture for the condensers, although his experience of it had been very favourable, the writer examined the other plans for surface condensation, in most of which the joints between the tubes and tube plates were made with vulcanized india-rubber;

but having understood that a chemical action took place between the copper of the tubes and the sulphur employed in preparing the india-rubber, and not being able to discover in the new plans any advantage over "Hall's" condenser, he adhered to this construction in the condensers of the "Mooltan." As regards the action of the vulcanized india-rubber on the copper tubes, the writer placed a piece of copper tube inside a piece of vulcanized india-rubber tube, and carefully washed and weighed the copper tube every month, and found a gradual decrease in its weight.

Mr. J. F. Spencer has often been alluded to in this work previously, in connection with the "cause and effect of surface condensers." He observed, during the discussion following Mr. Humphrys' paper, that he had also been working for many years at surface condensation, but with the opposite system of pumping the cold water through the interior of the condenser tubes, and condensing the steam on the outside; and he was glad now to learn the practical results of the working of "Hall's" condenser in a large ship, as described in the paper, the merits of that condenser having certainly not been fully appreciated.

In making a comparison between the two plans of surface condensers—the one with the condensing water outside the tubes and the steam inside, and the other with the steam outside and the water passing through the inside of the tubes—it was not necessary to consider either the space occupied by the condenser or the mode of making the joints at the ends of the tubes; because the space occupied depended entirely on the size of tube employed, and the same size might be adopted whether the water passed through the tubes or whether it passed outside; and the manner of making the joints by means of packing and screwed glands, as described in the paper, might be adopted for any plan of condenser. Setting these considerations aside, therefore, he considered an important practical difference between the two systems lay in the circumstance that, in order to examine a single tube of a condenser on the construction shown in the drawings, with the water outside the tubes, it was necessary to break a vacuum joint; and if such a joint were made again defectively at sea in a hurry, air would leak in, the vacuum in the condenser would be diminished, and the efficiency of working impaired. Whereas when the water was inside the tubes, all the ends of the tubes were accessible by simply breaking a water joint, which was a matter of little consequence; for if this joint were made defectively at sea, the only result would be a small outward leak of water out of the condenser, which would not affect the working of the engine in the slightest degree. In this respect, therefore, he thought a real practical advantage attended the plan of passing the water through the inside of the tubes.

A better distribution of water through the condenser was also obtained by the same plan of passing it through the tubes instead of outside. In the condenser shown in the drawings, with the water outside the tubes, he thought it would be almost impossible to pass the water thoroughly and equally over every portion of the condensing surface; whereas the water inside the tubes, by dividing the whole quantity of water into three or four currents distributed equally throughout the condenser, and by pro-

portioning the area of the tubes to that of the pump, the water might be driven over every portion of the condensing surface with almost complete uniformity. This he considered a very important point, and attributed to it much of the condensing power possessed by condensers having the water inside the tubes, with which he had obtained a condensation of 12 lbs. of water per hour per square foot of condensing surface, which he believed would be found greatly in excess of the general result. For judging of the efficiency of a condenser, the main point to be ascertained was, the weight of water condensed per square foot of condensing surface per hour, without any regard to either the nominal or the indicated horse p oer of the engine.

Mr. Spencer next imparted his experience in the adoption of india-rubber tube packing. "Out of 500 of these joints," he said, "he had but three cases of deterioration of the copper tube from the action of the india-rubber, and then even the fault lay with the defective fitting." Mr. Humphrys' experience as to price led him to state "that the cost of the tape packing—which he had adopted—was 16*s*. per thousand, ready coiled for use." Making further allusion to the proportions of the cooling surface, he said, "by keeping an uniform and constant flow of water over the surface of the tubes in the example he had described, 2½ square feet per indicated horse power was proved sufficient."

With reference to the position of the tubes in the condenser and the circuit of the water, it will be noticed that Mr. Spencer's remarks bore more reference to the "passage of the water amongst the tubes than the means of packing them." Mr. Cowper—taking part in the discussion—also threw some light on the matter by stating that "he had tried experiments to ascertain the effect of a current of water equal to 10 feet head: when running freely, it almost destroyed any air bubbles that might be formed. He thought, therefore, that the tubes ought to be vertical with the water passing through them, as the sectional area of the passage inside the tubes was less than that outside, also the velocity of the water would be greater over the surface. Moreover, in a forest of tubes there was a difficulty in getting any strong current of water into the middle of the forest, unless openings or spaces were purposely left between the tubes."

Now from these remarks it is obvious that Messrs. Spencer and Cowper were of the same opinion as to the course the water should circulate, viz., inside rather than amongst the tubes. Mr. Humphrys, however, had proved the efficiency of admitting water amongst the tubes, and the steam through them.

The author's experience in this matter has led him to conclude that there are advantages with the internal condensation not common with the outside mode, such as follows.

It is now a well known fact that with surface condensers the means for cleansing the tubes form the main consideration when designing them. Now when the steam surrounds the tubes, the upper half receives the solid accumulation, and thus the cooling power of the surface is greatly reduced. In extreme cases of incrustation, some of the spaces between the tubes are entirely closed by the refuse, and thus the passages for the condensed steam obstructed. It needs no

great enquiry as to the cause for this, particularly so on remembering that the steam itself is not a pure vapour, and that it is impregnated in the cylinder with the tallow, &c., although super-heating to a great extent mitigates the evil. The outer rows of the tubes may present a clean appearance, while those central are coated with refuse, and thus useless as conductors. To inspect the latter, the removal of the former is imperative, and therefore disarrangement the result. Now where the steam passes through the tubes, the refuse, of course, accumulate on the inner surface, and the removal of the coating is readily effected by a brush or suitable scraper. Each tube can thus be treated, and the certainty of cleansing the result.

The tube plates of a condenser always receive a portion of the solid matter left by the steam, and when the process of condensation is between the plates, inspection, even, is impossible without the removal of the tubes.

By the internal circuit now advocated, the plates are the first and last surfaces in contact, and thus the removal of any accumulation—to say nothing of inspection—is an easy process.

The modern practice by some of the leading firms of the day is to cause the steam to pass through an inclined set of tubes. The circulating water enters the chamber at a certain locality, and the discharge opening is opposite. In some instances a pipe is introduced —perforated—under the tubes, and thus the water is more effectually distributed.

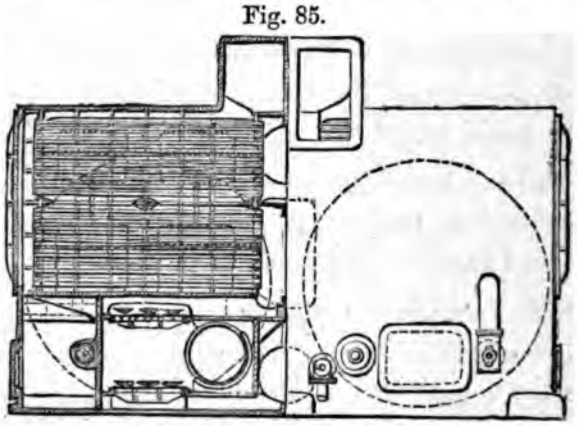
To enable a just and practical opinion as to the present practice, attention is now directed to the various arrangements.

SURFACE CONDENSERS FOR DIRECT ACTING ENGINES.

In common with the description relating to injection condensers, the present section commences with the details under notice, especially adapted for direct acting engines.

The firm of Messrs. J. Penn and Son have constructed and fitted in H.M.S. "Arethusa" the arrangement of condensers shown by

Fig. 85.



MESSRS. PENN'S SURFACE CONDENSER FOR DIRECT ACTING ENGINES.

Fig. 85—also shown in sectional plan and complete view in Plate 27.

The tubes are horizontally arranged above the pumps, the steam surrounding them within the compartment, the pumps being near the centre of the arrangement. The suction valves are below the barrel, at the side, and those for the discharge are directly above. The passage for the circulating water is thus. On the water entering at the central back opening, it is forced—by the pump's piston—through the central end of the first tier of tubes, and on reaching the opposite extremity it rises and enters the top set, the final discharge being through the upper opening at the back end of the structure.

The exhaust steam enters the compartment

at the front end, and after being condensed, falls to where the air pump suction valves are inverted. The discharge valves also are on the same level, the discharge chamber being at the side of the pumps.

It will be noticed that only two pumps are shown in the illustration, although both circulating and air pumps are requisite for each condenser. This is effected by using one end of each pump for circulating, and that opposite for the discharge of the condensed steam. Each pump is therefore single acting for the separate duties, but double acting in arrangement.

The pistons of the pumps are the ordinary disc kind, and the rods are connected directly to the trunk pistons of the engines. The condensing compartments are accessible by the removal of the doors—suitably arranged; while the tubes can be readily cleansed internally, or renewed when requisite.

The relative position of the tubes with the line of keel is parallel with the same—the means of cleansing and repairs being, therefore, fore and aft of the condensers.

As the proportions of this arrangement are worthy of attention, the following dimensions are introduced—the engines being 500 horse power nominal.

	Ft.	In.
Inside diameter of Tubes	0	$0\frac{7}{8}$
Outside ditto ditto	0	1
Length of ditto between Plates	6	$6\frac{1}{4}$
Number of ditto, 4,832.		
Diameter of Pumps, 2	1	8
Length of Stroke	3	6
Diameter of Pump Rod	0	3
Diameter of Suction Valves (air pump)	0	$8\frac{1}{2}$
Number of ditto ditto, 20.		
Diameter of Suction Valves (circulating pump)	0	$8\frac{3}{4}$
Number of ditto, 30.		
Diameter of Supply Water Pipe	1	8
Diameter of Discharge Water Pipe	1	8

Messrs. Penn do not confine their ideas of arrangement to the example illustrated. In some instances they adopt a vertical position for the tubes. The condensers are cylindrical, the tubes being concentrically arranged—to form a space, centrally, for the admission of the exhaust steam pipe. This latter detail is perforated at the extremity by parallel openings, to ensure a ready distribution. The steam surrounds the tubes while the water flows through the same—suitable compartments being formed above and below the plates.

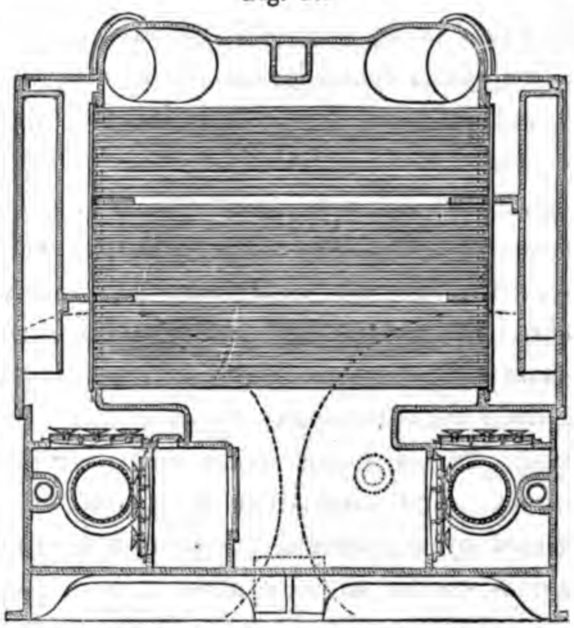
The air pumps are double acting, situated between the condensers—the suction and discharge valves being under and over the barrels. The circulation of the water is effected by centrifugal pumps and separate engines, similar to those alluded to in page 226. This arrangement has been fitted by the firm in H.M.S. "Bellerophon."

The arrangement of condenser and pumps, represented by Fig. 86—page 260—is that by Messrs. Dudgeon, fitted by them in the mail twin-screw steamer "Ruahine." The circulating pump is on the left-hand side, and the air pump on the right. The tubes are horizontally situated above the pump, in three clusters. The traverse of the water is through the tubes, and its circuit is thus: The vertical valves—suction—admit the water into the pump, and those for the discharge—horizontally situated—allow the flow into the first set of tubes. After the water has passed through to the other extremity, it rises to the second set, and from thence through the third set,

the final discharge being at the top of the compartment, at the back end. It is, perhaps, needful to add, that the supply and discharge pipes are fore and aft of the arrangement.

It will be noticed that two divisional portions are secured, between the corresponding

Fig. 86.



MESSRS. DUDGEON'S SURFACE CONDENSER FOR DIRECT ACTING ENGINES.

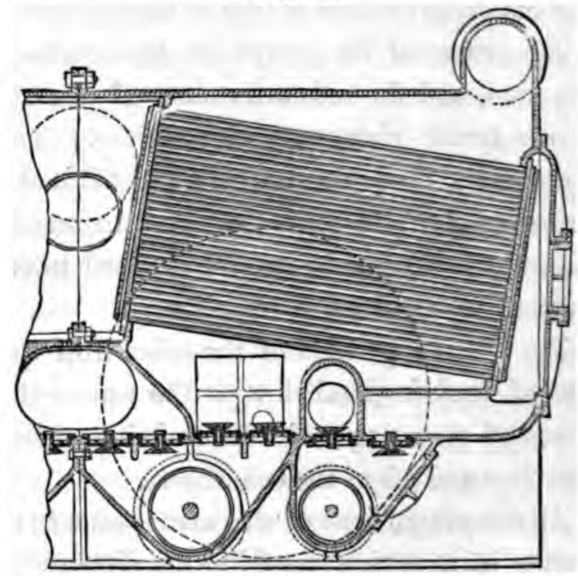
sets of tubes, to cause the forward and backward circuit alluded to. At each end of the tubes air chambers are formed, and thus the action of the pump is accelerated. The exhaust steam enters the tube compartment at the front end, above the tubes.

The air pump valves are similarly secured as those for the circulating pump. The action, therefore, being alike with each, the final discharge is effected at the back of the tube chamber.

The engines are arranged for twin-screw propulsion, of 350 horse power nominal collectively, and the principal dimensions of the condensing apparatus, for one pair of engines, are thus:—

	Ft.	In.
Inside diameter of Tubes	0	$0\frac{1}{2}$
Outside ditto ditto . . .	0	$0\frac{5}{8}$
Length between Plates	5	$7\frac{1}{2}$
Number of Tubes, 1,998.		
Diameter of Air Pump	0	11
Diameter of Circulating Pump . . .	0	11
Length of Stroke	2	0
Diameter of Pump Rod (air) . . .	0	$2\frac{1}{4}$
Ditto ditto (circulating) . .	0	$2\frac{1}{4}$
Diameter of Suction Valves (air pump) .	0	5
Number of ditto, 9.		
Diameter of Discharge ditto	0	5
Ditto ditto ditto (circulating pump)	0	5
Diameter of Supply Water Pipe . . .	0	9
Diameter of Discharge ditto	1	0

Fig. 87.



BURGH'S SURFACE CONDENSER FOR DIRECT ACTING ENGINES.

The arrangement illustrated by Fig. 87 is designed by the author, in connection with a pair of twin-screw engines, of 400 nominal horse power collectively, for an armour-plated steam ship.

The tubes are secured at an angle above the air and circulating pumps. The exhaust steam enters the compartment between the inner tube plates, and passes—right and left—through the tubes. The air pump—of the lesser diameter—drains the compartment at the outer extre-

mities of the tubes, and thus a continuous and almost direct current is effected. The suction and discharge valves are above the pump on the same level—the passage above the latter communicates with a tank—thereby a free action is maintained. The circulating pump —of the greater diameter—is at the side of the air pump, both suction and discharge valves being similarly situated for each purpose. The circulating water enters through the oval opening at the back, and, after passing amongst the tubes, it is discharged through the pipes at the top of the compartments.

The main effect attained with this arrangement is a duplication of the entire working details for each engine—with single pipes for the supply water and exhaust steam common to both engines. By this a saving of connections results, also the weight of material and space occupied in the hull.

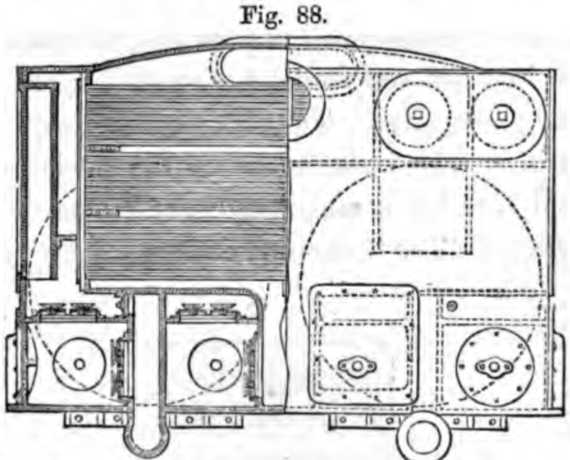
The doors for access to the several compartments are all suitably arranged, and on the main or large doors those of a small size are secured, the removal of which being for inspection only. It is worthy of remark, that even by a reverse situation for the pumps and the steam surrounding the tubes, the arrangement will be but little affected—at least the tubes can remain as illustrated; the circulating water entering at the top of the central compartment, and the condensed steam being discharged through the central opening below, with the valves reversed also.

The principal dimensions of the pumps, valves, &c., for one pair of the engines alluded to are thus:—

	Ft.	In.
Inside diameter of Tubes	0	0⅝
Outside ditto ditto	0	0¾
Length	5	0
Number of Tubes, 4,988.		
Diameter of Air Pumps, 2	0	10
Diameter of Circulating Pumps, 2 . .	1	1
Length of Stroke	2	6
Diameter of Pump Rod (air) . . .	0	1½
Ditto ditto (circulating) .	0	2⅛
Diameter of Suction Valves (air pump) .	0	6
Number of ditto, 12.		
Ditto ditto Discharge Valves, 12.		
Diameter of Suction Valves (circulating pump)	0	6½
Number of ditto, 24.		
Ditto ditto Discharge Valves, 24.		
Diameter of Supply Water Pipe . .	1	5
Diameter of Discharge Pipes, 2 . .	1	0

To be noticed next, as a novel arrangement, that depicted by Fig. 88 is introduced. This is

Fig. 88.



MESSRS. WATT'S SURFACE CONDENSER FOR DIRECT ACTING ENGINES.

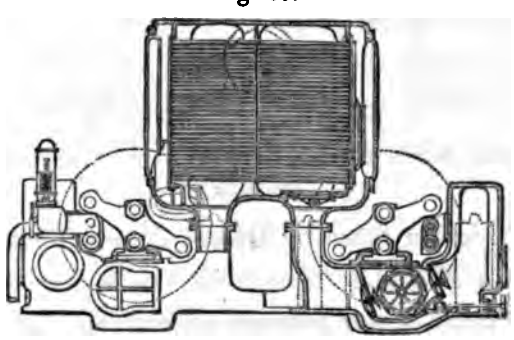
a condenser fitted by Messrs. James Watt and Co. in H.M.S.S. "Research," with engines of 200 horse power nominal collectively. It will be noticed the principle of the arrangement is precisely as that depicted by Fig. 86—page 260 —the exception in arrangement being only that duplicate "air and circulating pumps" are introduced in the present case. As the locality of the detail is nearly the same in each example, it will, for the present purpose, be only necessary to allude to the dimensions in connection with the example under notice.

	Ft.	In.
Inside Diameter of Tubes	0	0⅜ 1/32
Outside ditto	0	0½
Length between Plates	7	1
Number of Tubes, 2,506.		
Diameter of Air Pumps, 2 . . .	0	10
Diameter of Circulating Pumps, 2 . .	0	10
Length of Stroke	2	0
Diameter of Pump Rod (air) . . .	0	2¾
Ditto ditto (circulating) .	0	2¾
Diameter of Suction Valves . . .	0	5½
Ditto of Discharge ditto . . .	0	5½
Number of each Set, 12.		
Diameter of Supply Water Pipes, 2 .	0	6½
Diameter of Discharge Pipe . . .	0	10

SURFACE CONDENSERS FOR RETURN ACTION ENGINES.

The remarks given in page 249, in relation to injection condensers, very nearly apply in the present case. The piston and connecting rods in each instance cut up the condenser, and thus the arrangement is, of course, duly effected. The illustration depicted by Fig. 89

Fig. 89.



MESSRS. NAPIER'S SURFACE CONDENSER FOR RETURN ACTION ENGINES.

is the latest example by Messrs. R. Napier and Sons, for a pair of engines of 300 nominal horse power collectively. The tubes are arranged above the pumps in one compartment, the steam surrounding the tubes and the water passing through them. The circulating and air pumps are situated directly below the guide channels, each being worked by the opposite steam pistons. The valves are rectangular, located at an angle on each side of the barrel —the discharge chambers being outside the piston rods of each engine. The circulating water is intercepted at three points during its traverse, and thus a return action within the tubes is effected, similar to that described in page 259, in relation to Fig. 86—page 260.

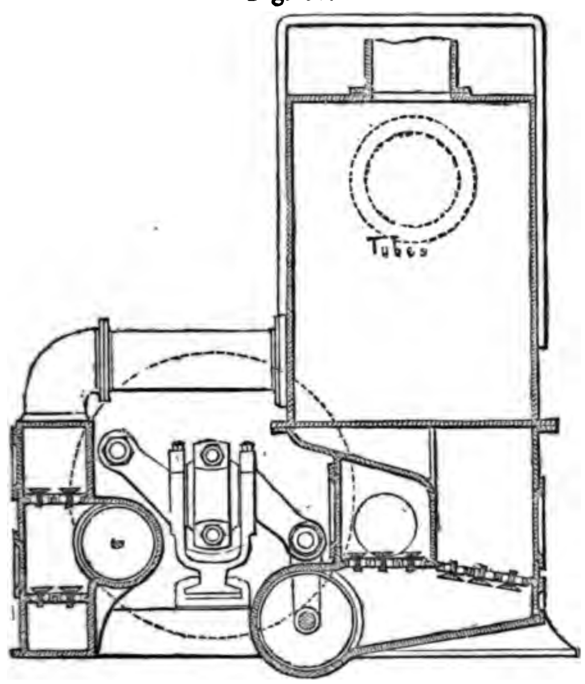
The practice of Messrs. Napier being worthy of attention, the following dimensions in relation to Fig. 89 are introduced:—

	Ft.	In.
Inside Diameter of Tubes	0	0⅝ full.
Outside ditto ditto	0	0¾
Length of ditto between Plates . .	6	3
Number of Tubes, 3,923.		
Diameter of Air Pump	1	4
Ditto of Circulating Pump	1	4
Length of Stroke	2	6
Diameter of Pump Rod (air) . .	0	2¾
Ditto ditto (circulating) .	0	2¾
Sizes of Valve Gratings (27½ × 5 ins.)		
Number of Valves to each Pump, 4.		
Diameter of Supply Water Pipe . .	1	4
Diameter of Discharge Pipe . . .	1	4

Messrs. Maudslay, Sons, and Field have been mentioned as originators of two kinds of injection condensers; they also have produced two arrangements of the surface type. Fig. 90 is a sectional elevation of their condenser, having the tubes inclined athwart-ships. The air pump is directly under the inner piston rod, and derives its motion from an arm connected to the same. The suction valves are inverted below the tubes, and those for the discharge are at the side of the barrel above the same. The exhaust steam enters the tubes at the front end and passes to the back, the final discharge being into the chamber between the pump and the tubes. Next, it will be

noticed that the "circulating pump" and its valves are separate from the condenser and air

Fig. 90.

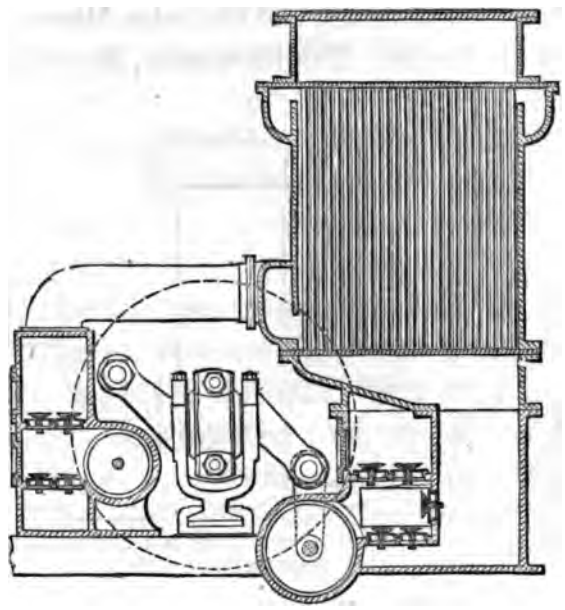


MESSRS. MAUDSLAY'S SURFACE CONDENSER FOR RETURN ACTION ENGINES.

pump — unlike any previous arrangement alluded to. This pump is under the outer piston rod, and derives its motion from the steam piston. Both suction and discharge valves are at the side, under and over the pump. The circulating water is conveyed from the pump into the lower part of the condenser by the pipes above the piston rods, and the discharge from the compartment is at the roof. The pumps are double acting, with ordinary disc valves, and doors are suitably arranged where required.

Similar in principle and almost in arrangement, the example represented by Fig. 91 is another production by the firm in question. In this case the tubes are vertical, the pumps being located as described before. The exhaust steam enters the compartment above the tubes at the front end, and falls direct to the space under

Fig. 91.



MESSRS. MAUDSLAY'S SURFACE CONDENSER FOR RETURN ACTION ENGINES.

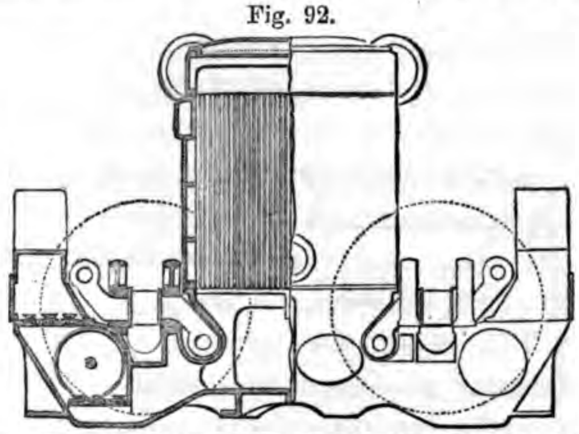
the suction valves,—the latter being at the side of and below the pump. The description of the remaining portion of the action and detail, in allusion to Fig. 90, applies in the present case; further notice is, therefore, unnecessary.

Messrs. Maudslay have fitted several ships lately with these condensers, and for a pair of engines of 300 horse power nominal, the proportions of the details under notice are thus:—

	Ft.	In.
Inside Diameter of Tubes	0	0½
Outside ditto	0	0⅝
Length	6	5
Number of Tubes, 4,872.		
Diameter of Air Pumps, 2 . . .	1	3⅞
Ditto of Circulating Pumps, 2 . .	0	10½
Length of Stroke	2	6
Diameter of Pump Rod (air) . . .	0	2
Ditto ditto (circulating) . .	0	1¾
Diameter of Supply Water Pipes, 2 . .	0	11½
Diameter of Discharge Pipes, 2 . .	1	2½

Besides the firm just alluded to, others have adopted the vertical position for the tubes and inside condensation, amongst whom are Messrs. Ravenhill, Hodgson, and Co.; also Messrs. J. and G. Rennie. The illustration, Fig. 92, is

Fig. 92.



MESSRS. RENNIE'S SURFACE CONDENSER FOR RETURN ACTION ENGINES.

an example by the latter firm, of late production, for a pair of engines of 350 horse power nominal collectively. The exhaust steam enters the tubes at the top extremities, and passes direct down to the air pump—the latter being located directly below the outer piston rod. The circulating water is drawn through the compartment, entering at the top, and discharging at the bottom—rather than beingforced through the compartment. Both circulating and air pumps—one of each—are worked by the respective steam pistons. The tubes, $\frac{9}{16}$in. in diameter outside, are 5 feet long, and the total cooling surface is 7000 square feet, or 20 square feet nominal horse power. The pumps are each 1 foot 9 inches in diameter, having a stroke of 3 feet.

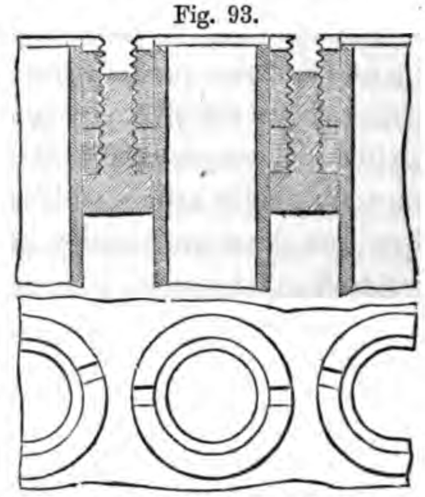
In the event of a fracture of the tubes, or the requisition of the injection system in the surface condensers alluded to, suitable valve and other requisite details are appended, as illustrated, for that purpose. For example, in page 260, the arrangement—Fig. 86—there alluded to, has a perforated hollow passage below the roof to receive the injection water, and the circulating pump can be rendered effective as an air pump by the opening of the plate valve above and the removal of the door at the side. These remarks apply also very nearly to the provisions introduced in the arrangement represented by Figs. 88 and 89—pages 261 and 262. It may be added, as a conclusion to this question, that it is more practicable to reverse the system with outside tubular condensation than when the inside surface is adopted.

TUBE PACKING.

The different modes introduced for making a perfect joint around the tubes of surface condensers have been plentiful. Practice in this matter has now, however, almost settled which is the best means, and consequently to those only will allusion be given.

Mr. Humphrys' experience in the question is quoted in page 255, and the illustration below—Fig. 93—is the type he has alluded to.

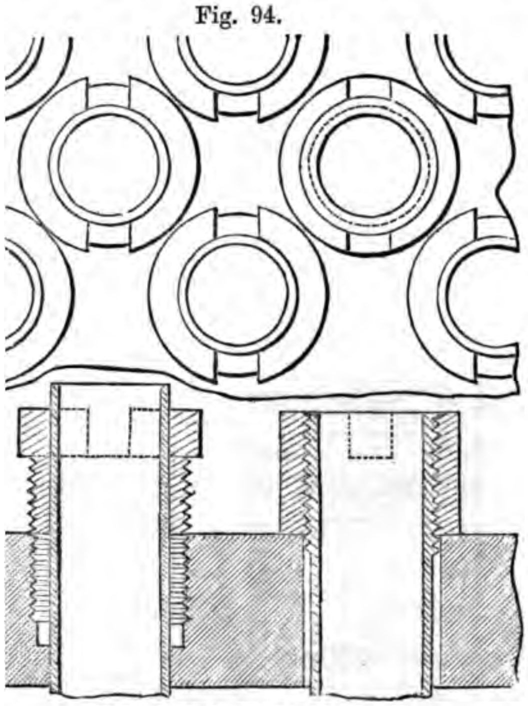
Fig. 93.



MESSRS. HUMPHRYS' (HALL'S) GLAND TUBE PACKING.

glands, it will be noticed, have no pro- ns above the screwed portion, by which ion the process of screwing them down loubtless in time disturb the form. Now, viate this, we sometimes use the glands, own by Fig. 94; and we also use certain

Fig. 94.

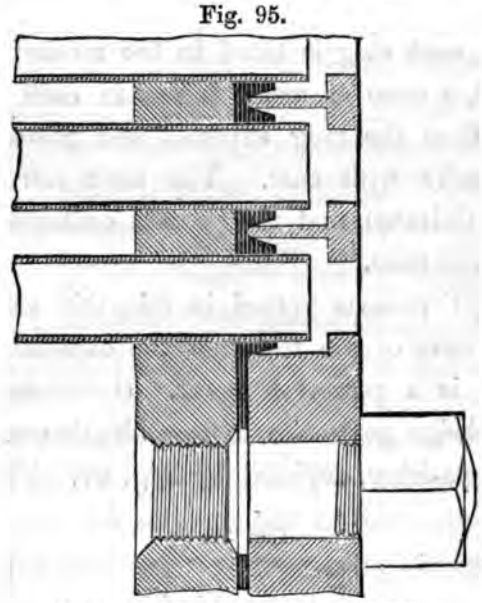


BURGH'S GLAND TUBE PACKING.

as stays, to resist the pressure between lates when the water circulates amongst ibes.

though glands and packing are cer- in effect, authorities on the subject, in instances, doubt their adoption as a sity. Mr. J. F. Spencer, for instance, s, in Plates 5 and 6, that two washers of -rubber surrounding each tube is *alone* ient to make a perfect joint, the vacuum een the plates causing the requisite com- ion. Other engineers prefer a plate or of india-rubber perforated to suit the ion of the tubes, and a second, or loose plate of metal compresses the packing around them. This mode is shown by Fig. 95, being

Fig. 95.

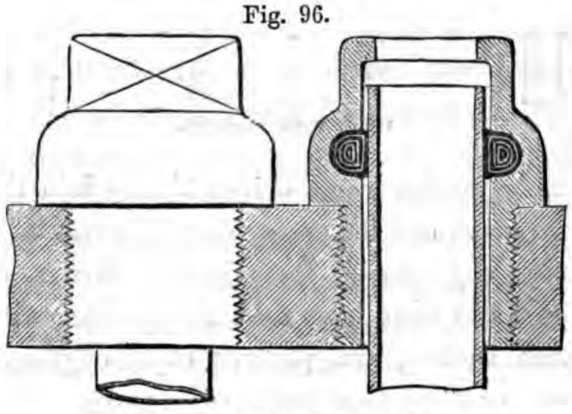


INDIA-RUBBER PLATE TUBE PACKING.

adopted by some of the English and Scotch firms of distinction.

The compression of the packing, it will be noticed, is the main consideration, and this is accomplished in a novel manner by Mr. J. G. Winton, illustrated by Fig. 96. The pack-

Fig. 96.



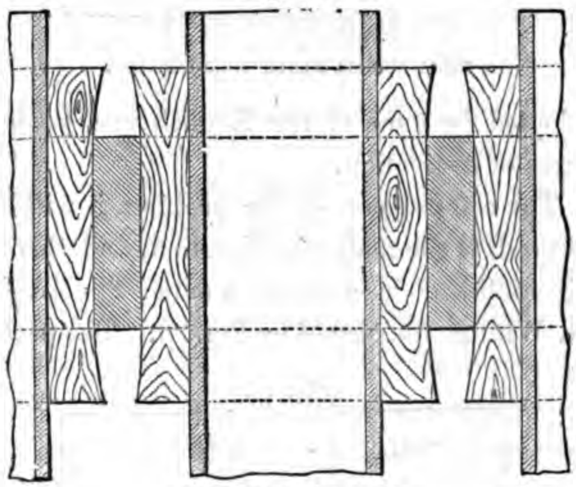
WINTON'S INDIA-RUBBER RING PACKING.

ing for each tube is a ring—circular in sec- tion—of india-rubber, which is compressed by its elasticity to fit the tube, as shown. The

recess formed in the gland prevents a lateral disturbance, while the tube can shift by contraction and expansion at will. To pack the tubes, each ring is fitted in the recess in the gland, a cone of metal is put in each tube, and thus the ring expands and passes the extremity with ease. The cone can then be withdrawn, and the process continued for the next tube.

Most persons versed in scientific matters are aware of the fact that the expansion of wood is a powerful agent. Doubtless this knowledge gave rise to the adoption of the wood packing, depicted by Fig. 97. This is

Fig. 97.



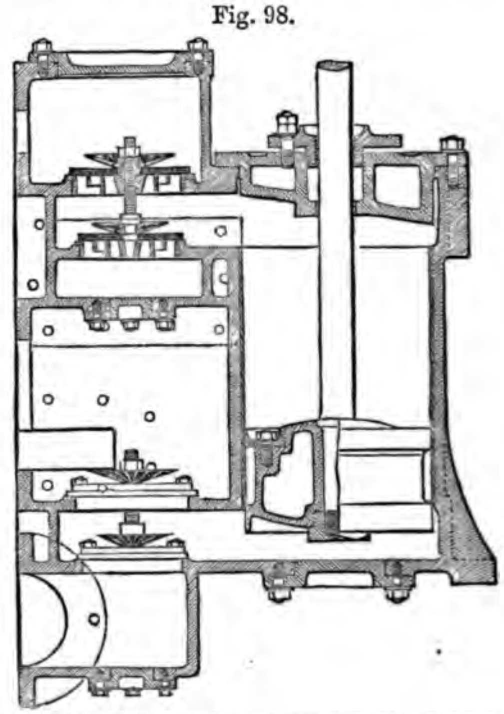
WOOD TUBE PACKING.

simply wooden rings driven tightly into the tube plate, and the expansion of the vegetable causes a joint with the mineral. Messrs. J. Penn and Sons have used this packing with much success, and perhaps other engineers have, or will follow in the same course. The example illustrated is rather an extreme size. Messrs. Penn's practice for tubes $\frac{3}{4}$ inch outside diameter is: the packing to be $1\frac{1}{8}$ inch, and the hole in the plate $\frac{15}{16}$ inches in diameter.

AIR PUMPS.

When an engineer has to design an air pump, he has two functions at his command, viz., single or double action, with vertical or horizontal positions. The double action is now become an universal mode, hence the single type will not be alluded to. The vertical position having been first developed, attention to it has the precedence, by the illustration, Fig. 98, which is a sectional

Fig. 98.



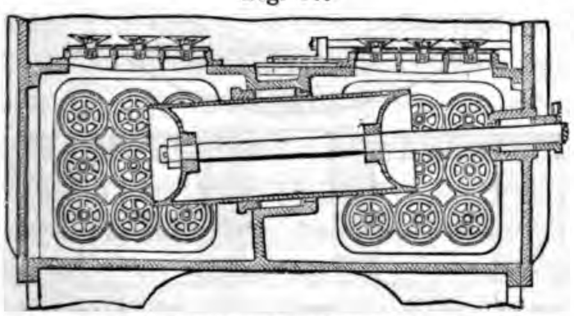
SPENCER'S VERTICAL SINGLE ACTING AIR AND CIRCULATING PUMP.

elevation of a vertical single acting air and circulating pump, by Mr. J. F. Spencer. The suction valves are at the extremity of the barrel, at the top, and below the same at the bottom, the discharge valves being in reverse positions. By this arrangement the piston, during its up-stroke, causes the lowest valve to rise simultaneously for the suction, and a similar effect is caused for the top valve by

the discharge. The down stroke of the piston produces a reverse action; the relative valves are closed, and those intermediately situated opened, the discharge being therefore through the upper valve-seating in either case. This position of the details admits a continual flow of the water and air, but the cubic contents of the pump are in proportion for circulating purposes, rather than for air.

The illustration, Fig. 99, is a sectional elevation of the pump and valves of the

Fig. 99.

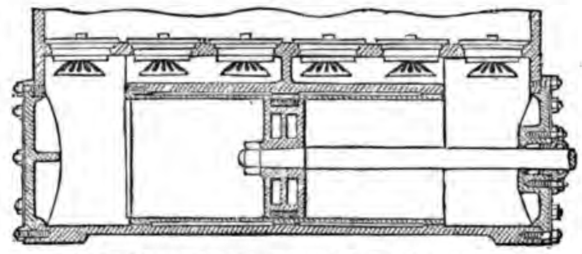


DUDGEON'S HORIZONTAL PLUNGER AIR PUMP.

condenser represented by Fig. 86—page 260. The position of the valves has already been described, and it only remains therefore to allude to the piston. It is preferred in this case to adopt a plunger, in the place of the disc kind. The barrel is therefore dispensed with, and the packing is an ordinary gland and stuffing box, the former being adjusted by studs and nuts. One of the advantages with this over the disc piston is that the plunger displaces the whole of the water at each stroke, when the space is constructed for that purpose. In the illustration under notice the plunger is inclined, and the space below is enlarged to receive the vertical valve plate.

Next to be noticed is the horizontal disc piston pump, represented by Fig. 100. This is the circulating pump belonging to the condenser shown by Fig. 87, in page 260. The suction valves are only seen in this view, inverted, above

Fig. 100.



HORIZONTAL DISC PISTON CIRCULATING PUMP.

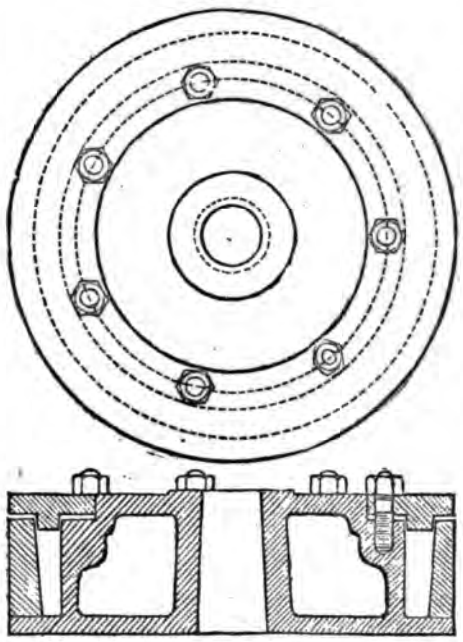
the pump, the discharge valves being, as already stated, on the same level. The main attainment with this arrangement is, the barrel can be at the base of the structure, and the length of the spaces beyond the barrel is in proportion to the area of the piston. Many of the leading firms adopt this arrangement, and the injection condenser, shown by Figs. 75 and 78—pages 247 and 249—are similarly fitted.

PUMP PISTONS.

The details now under notice have had due attention from the engineers interested in their efficiency. For air pumps the pistons are of two kinds, gasket and metallic. The former is shown half in section in the pump represented by Fig. 98—page 266—and the latter in plan and section by Fig. 101, in page 268. In each example the compression of the packing produces the perfect contact requisite with the barrel. With the gasket type the material itself is in immediate contact; but with the metallic kind a ring intervenes between the packing and the barrel, and thus the wear of the material in constant action is reduced. The evil here obviated is

common with the plunger piston, unless a metallic ring encompasses the same.

Fig. 101.

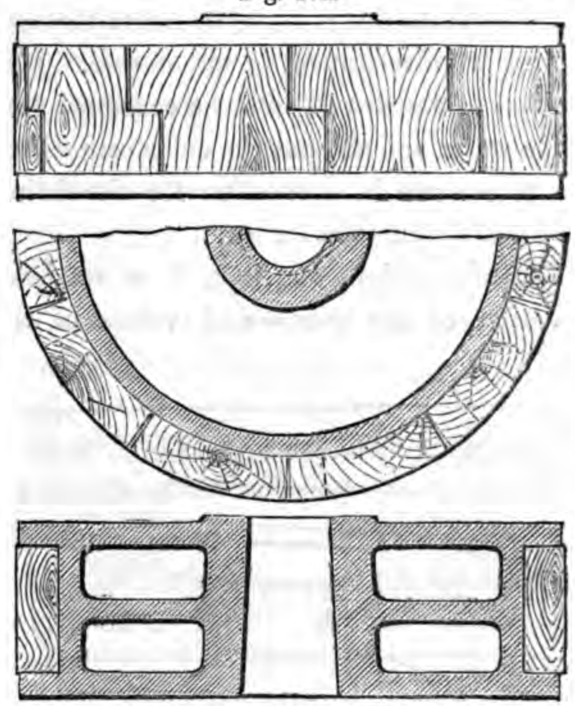


METALLIC PISTON.

Again glancing at the example shown by Fig. 101, it will be seen that the ring is of unequal thickness, the division for expansion being at the thinnest part. The face ring is secured or adjusted by studs and nuts, and the looseness of the latter is prevented by check nuts, side studs, or a stop ring, neither of which are shown, as either can be adopted with similar effect. It is almost obvious from the section that the packing, when compressed, causes the ring to expand, and thus a perfect contact is attained, as already alluded to. The type of piston just described has been often adopted for circulating pumps of surface condensers.

Messrs. Penn, some years ago, proved the utility of lignum-vitæ as a bearing for a metallic surface revolving in water. Now this knowledge produced the piston packing for circulating pumps, as shown by Fig. 102.

Fig. 102.



BURGH'S WOOD PACKING FOR PISTON.

This has been adopted with much success by Messrs. Maudslay and other engineers who have constructed surface condensers.

We have often used it, and also with the condenser represented in page 260 by Fig. 87. The illustrative view above—Fig. 102—is the piston of one of the circulating pumps, and is shown in sectional and complete views. The body of the piston is of ordinary design; the packing is composed of curved strips of lignum-vitæ, each being recessed where in connection. This is shown in the complete elevation, and the plan being in section, the length of the strips are apparent, the sectional elevation showing the thickness of the material. This mode of dividing, or rather, arranging the packing, is practically good; but, doubtless,

other means, similar in principle, will produce the desired effect. The main question to be solved is the proportion of the strips to the circumference of the piston, so that the expansion of the wood shall not fracture the barrel, and at the same time be in good working condition.

AIR-PUMP VALVES, AND MODES OF SECURING THEM.

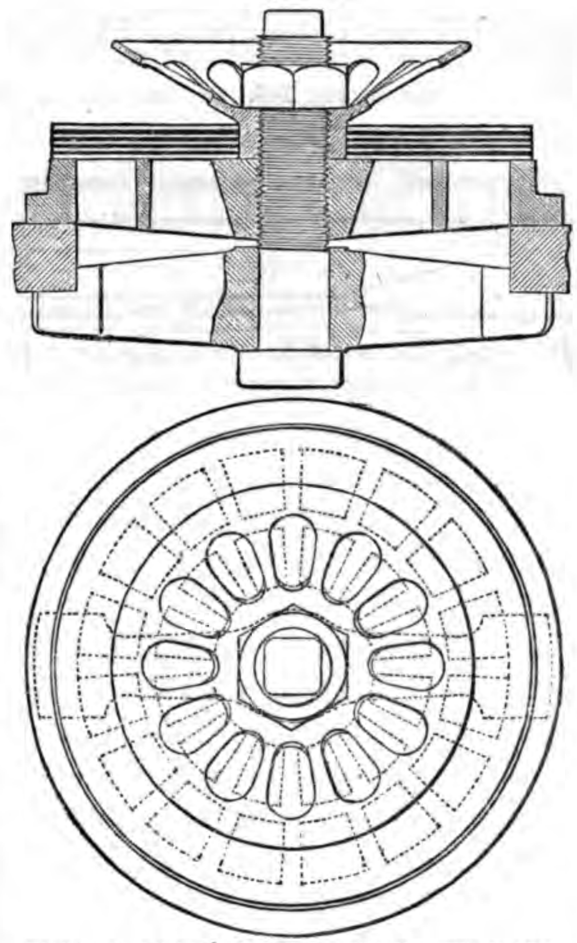
The air pump valve, as well known, is composed of india-rubber, acting on a seating perforated according to the experience of the designer. To prevent the valve from rising too high, and its non-return, a guard is introduced. From this it is certain that three details have to be considered, viz., the valve, seating, and guard.

The example shown by Fig. 103 is the mode adopted by Messrs. Maudslay. The guard and seating are perforated discs of suitable forms, the valve being located between them. The seating and guard are secured by a single bolt and nut. The bolt being screwed through the seating and guard, and loose through the cross-bar below, a certain connection is insured. The bolt is tightened by the square extremity above the nut.

For inverted valves, the firm under notice adopt a similar connection, represented by Fig. 104; the difference in this and the previous example being that the guard is flat instead of angular.

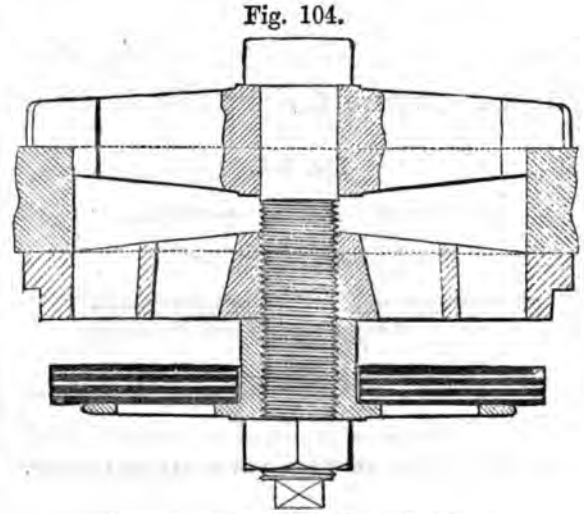
Some firms prefer a plate, secured in the condenser, perforated according to the number of valves requisite. A portion of a plate thus formed is represented by Fig. 105—page 270. The plate is secured by ordinary studs and nuts, at a pitch in proportion to their diameter. The

Fig. 103.



MESSRS. MAUDSLAY'S MODE OF SECURING THE VALVE-SEATING AND GUARD.

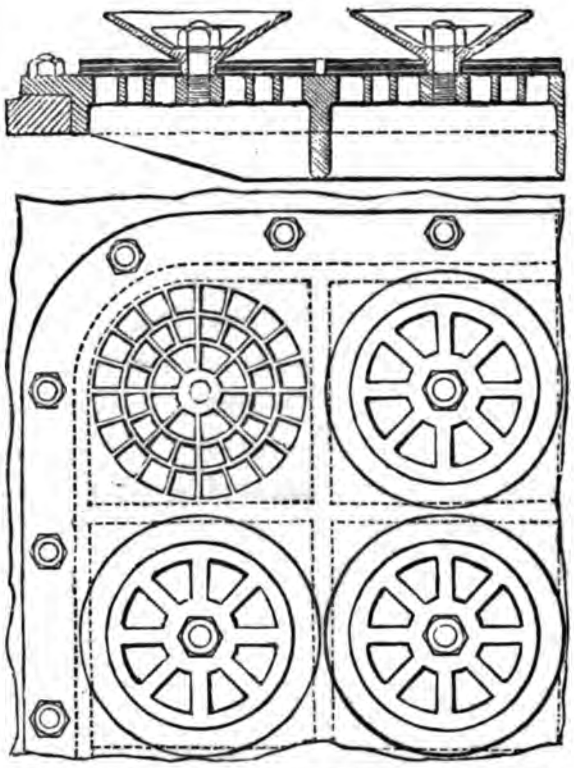
Fig. 104.



MESSRS. MAUDSLAY'S INVERTED VALVE

seatings are arranged at right angles with each other, and ribs cast between them add strength

Fig. 105.

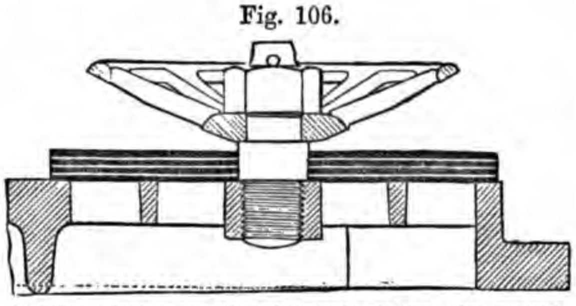


MESSRS. PENN'S MULTIVALVULAR PLATE AND VALVES.

to the portions intervening. The guards are secured by studs and nuts, and the valve surrounds the projection on the guard.

The more universal mode is, as that shown by Fig. 106, where the projection is omitted,

Fig. 106.

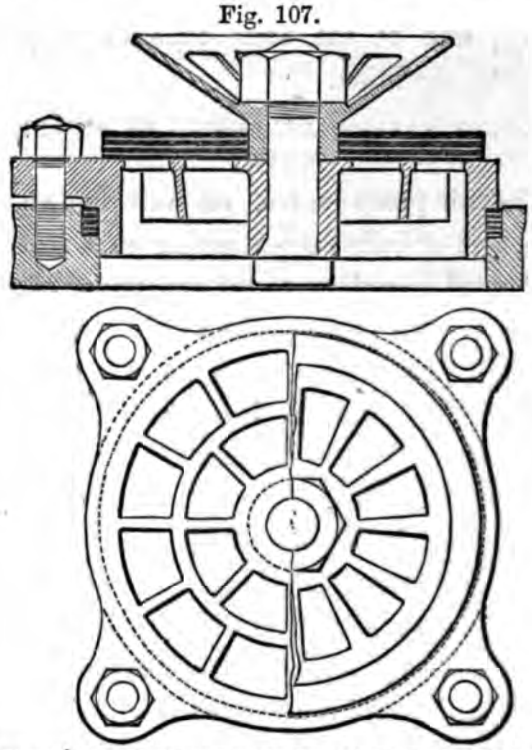


GENERAL MODE OF SECURING THE VALVE AND GUARD.

and a collar on the stud accomplishes the means for sustaining the valve, as well as supporting the guard.

Our practice in these matters is shown by Fig. 107, where each seating is a separate

Fig. 107.



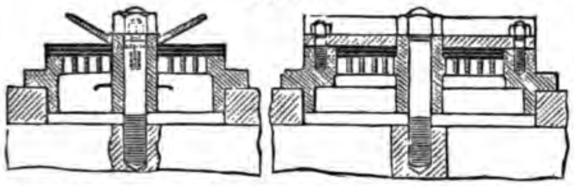
BURGH'S MODE OF SECURING THE VALVE-SEATING AND GUARD, INVENTED IN 1859.

perforated disc, with three or four projections, for the purpose of securing, by studs and nuts, to the condenser. The joint is made by an india-rubber ring, fitting in a recess cast for the purpose. The guard has a projection bearing on the seating, and an ordinary bolt and nut complete the connection. With this arrangement only one pattern is requisite, of small size,—in proportion to the plate kind—and the only portion of the seating requiring surfacing is the upper side. Thus labour is economised.

Twin-flap or butterfly valves are sometimes introduced with much success. The firm most

partial to this kind is Messrs. Humphrys and Tennant, and to render their practice apparent, the illustration, Fig. 108, must now be

Fig. 108.

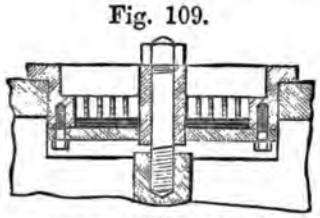


MESSRS. HUMPHRYS' MODE OF SECURING THE VALVE-SEATING AND GUARD.

alluded to. The plan of the seating is rectangular; the raised portion above the seating—formed with the same—passes through both valve and guard. The seating is secured by a single stud and nut, centrally secured, and two studs and nuts secure the guard and valve at the extremities of the guard. The sections illustrate longitudinal and transverse views of the detail in question, therefore further description is scarcely requisite.

The inverted valve is nearly similar in design and connection as the previous example. This will be understood by alluding to the Fig. 109, which is a transverse section. The

Fig. 109.



MESSRS. HUMPHRYS' INVERTED VALVE.

joint of the seating is above the valve, and a single stud and nut ensures the connection. The longitudinal view is as that in Fig. 108, the position of the seating flange excepted. Each seating is rectangular, with straight sides and ends, and being in contact with each other, multiple support is the result.

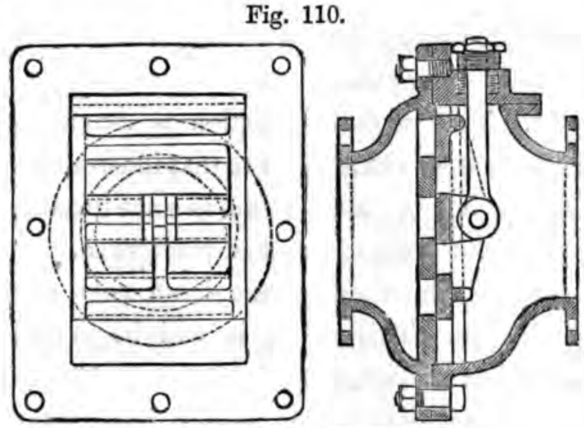
Messrs. Napier also adopt this type of valve, as shown in Fig. 89—page 262—the seating being secured by studs and nuts, and the guard and valve, centrally, in the same manner.

INJECTION AND SNIFTING VALVES AND SPRAY PIPES.

The injection valve, it need scarcely be stated, regulates the flow of the water into the condenser; because, as the pressure is but slight, the arrangement of the details becomes a simple matter. The general practice by the majority of the leading firms is, a plate of metal for the valve, which closes or opens an aperture of less dimensions, the valve being enclosed in a casing of suitable form. The motion for the valve is attained by a lever attached to the rod, and thus a sliding action can be produced when required.

The illustration, Fig. 110, is an injection valve of the gridiron type, some time since

Fig. 110.



GRIDIRON INJECTION VALVE.

introduced by marine engineers to reduce the stroke of the valve requisite for the admission and stoppage of the injection water; in this way—that it is obvious if the area of the requisite passage is subdivided into ports, and

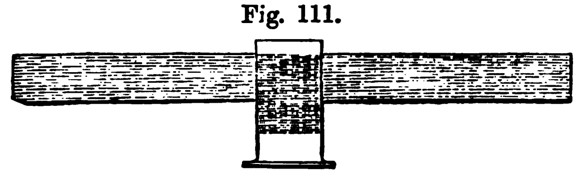
the valve to correspond with a lap, the flow of the water can be checked or regulated with the most sensitive manipulation; while it may be added, in passing, that the plate valve is of precisely similar arrangement, excepting the subdivision, the valve then being solid, and the seating a single passage. The position of the branch for the connection of the "Kingston" pipe is either at the front or bottom of the casing, each locality being determined by the arrangement in the hull. For example, Messrs. Penn prefer the base of the casing for the connection, while Messrs. Maudslay attach the pipe at the front, or similar to Fig. 110. Messrs. Napier also prefer the latter practice. Messrs. Dudgeon, on the contrary, decide as Messrs. Penn, each position being therefore alike in effect.

The details under notice relate to the sea injection; but a supplementary valve is added to the condenser in most cases, termed a bilge injection valve. This addition is used only when the valve proper is useless, or more often when the air pump is required to empty the bilge. The valve in question is usually an ordinary stop-cock, similar to that illustrated by Fig. 65, in page 235; while in some instances a plate slide valve is used in preference.

The snifting valve is the detail next to be alluded to. This valve, it will be remembered, has had ample notice in pages 236 and 237, in relation to its utility with paddle engines. As the principle of condensation in both types of engines is alike, it remains but to state that the remarks and illustrations there alluded to apply in the present case. This is rendered fully obvious, on remembering that the use of the valve in question is to permit the escape of the air and water, and also prevent the return of the same into the condenser.

The spray pipes, in connection with the injection valves, are secured beyond them, within the condenser. The illustrations Fig. 111, is the practice of Messrs. Penn, being

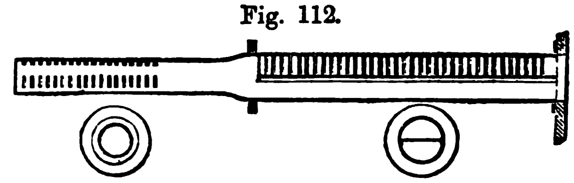
Fig. 111.



MESSRS. PENN'S WATER SPRAY PIPE.

a perforated pipe with a central branch for the admission of the water. Messrs. Maudslay adopt the shape illustrated by Fig. 112. This

Fig. 112.



MESSRS. MAUDSLAY'S WATER SPRAY PIPE.

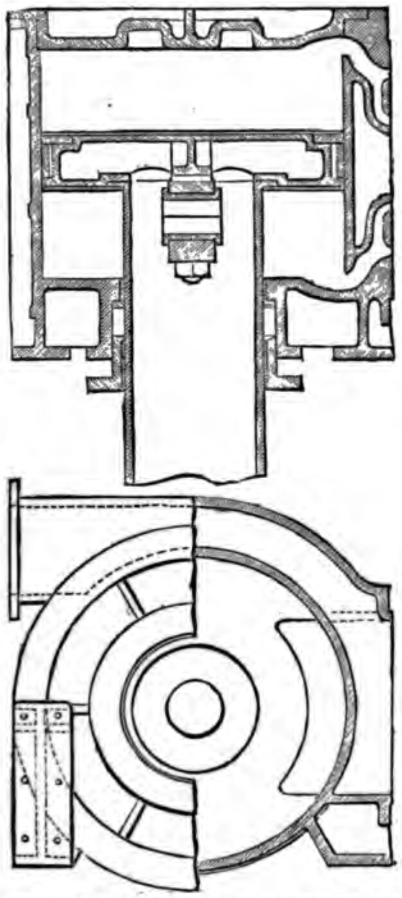
pipe is of unequal diameters, the larger area being twice that of the smaller. The first portion of the pipe is divided centrally by a plate throughout its length,—shown also transversely. It will be noticed also that the perforations are on the upper side only. The portion beyond the flange connecting the two pipes is perforated equally on the surface. The utility of the division plate is, that the water shall be equally distributed throughout the pipe's length. This, it may be added, is also accomplished by Messrs. Penn, by the right and left passages for the water, or by the central opening for its admission.

STEAM CYLINDERS.

In common with the description of the various screw engines and their condensers,

this section will commence with a notice of the cylinders and pistons for trunk engines. Messrs. J. and G. Rennie arrange the detail under notice as depicted by Fig. 113. The

Fig. 113.



MESSRS. RENNIE'S CYLINDER AND SINGLE TRUNK PISTON.

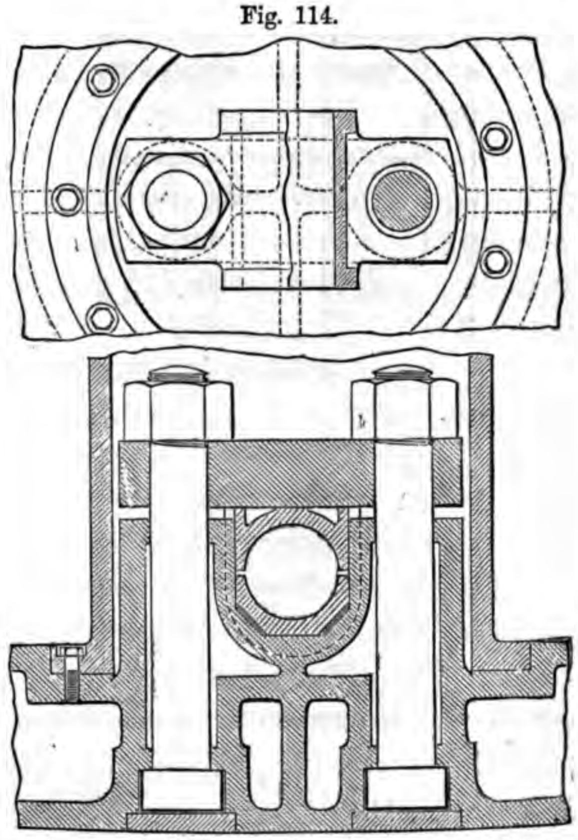
plan, being in section, shows the piston at half stroke; and the elevation, half in section and half complete, with the piston omitted, represents the form of the cylinder and the direction of the exhaust passage. The latter, it will be noticed, passes over the body of the cylinder, where it connects directly with the condenser,—as shown in plan by Fig. 4, page 44.

The illustration, Fig. 113, is that of a cylinder of 100 horse power nominal, the principal dimensions being as follows:—

	Ft.	In.
Diameter of Cylinder	3	7
Diameter of Trunk	1	8
Length of Stroke	2	0
Length of Connecting Rod	5	0
Depth of Piston	0	8

The mode of adjusting the brasses for the connecting rod at the piston end of the trunk has been a matter of the deepest thought by those who advocate trunk engines. The Messrs. Rennie have two modes of overcoming the difficulties, one relating to a single-end connecting rod, and the other for the double

Fig. 114.



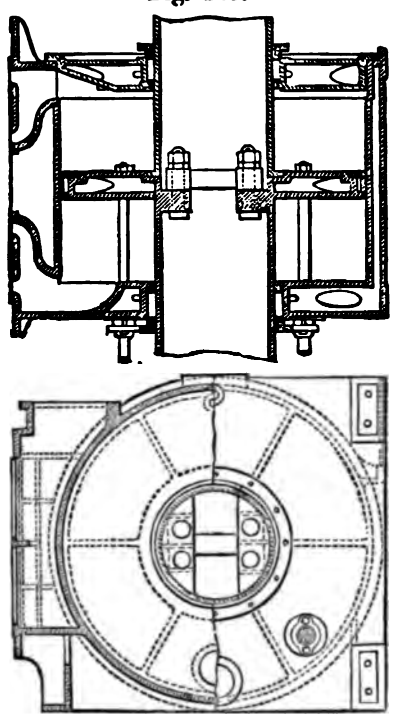
MESSRS. RENNIE'S MODE OF ADJUSTING THE BRASSES FOR THE CONNECTING ROD PIN.

or forked-end type. The arrangement, shown by Fig. 114, is a single plummer-block, cast

with the piston, the trunk being secured by a flange, studs, and nuts. The connecting-rod end is forked, and clasps each link of the brasses, thus preventing lateral movement: the remaining portion is obvious from the drawing alluded to. When a single-end connecting rod is introduced, the cross-head pin is secured at each end of the bearing to portions suitably formed with the piston, and the brasses are adjusted, beyond the trunk, by a rod inserted centrally through the connecting rod.

Messrs. Penn prefer the double trunk arrangement, illustrated by Fig. 115. The front

Fig. 115.



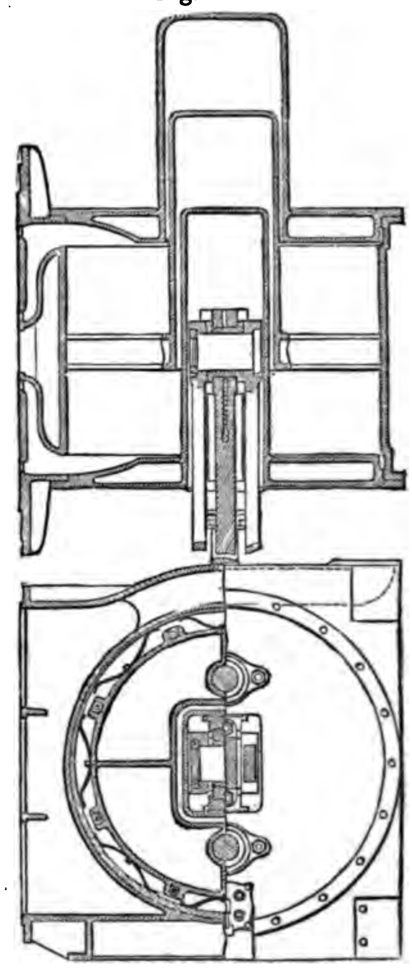
MESSRS. PENN'S CYLINDER AND DOUBLE TRUNK PISTON.

trunk is cast with the piston, but the back trunk is secured by studs and nuts. The cross-head is also secured by bolts and nuts, to portions formed within the front trunk at the piston end. The exhaust passage—seen in the elevation—terminates at the side of the cylinder, so that its cooling effects are obviated as much as possible. For expansive engines of 500 nominal horse power collectively, the proportions of the cylinders and trunks are thus:—

	Ft.	In.
Diameter of Cylinder	7	2½
Diameter of Trunk	2	9
Length of Stroke	3	6
Length of Connecting Rod . . .	10	3
Depth of Piston	0	7¾

The friction of the trunk engine is its bane, and as an antidote, we did, in the year

Fig. 116.



BURGH'S PATENT CYLINDER AND ANTI-FRICTION TRUNKS.

1859, invent and patent the arrangement illustrated by Fig. 116. The front cover of the

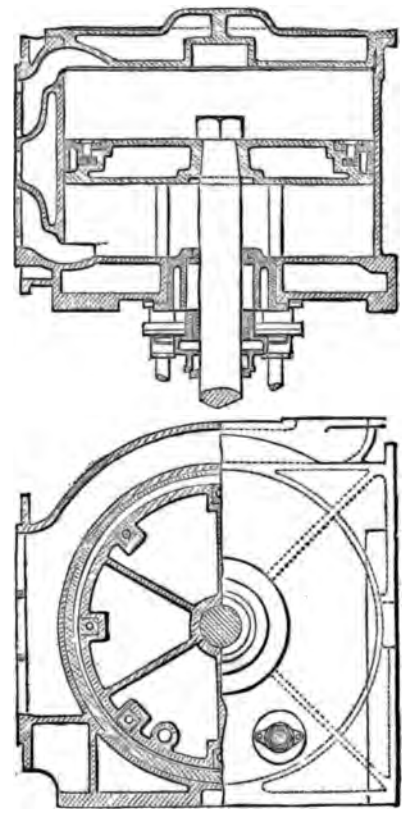
cylinder has cast with it, or secured, a trunk equal in length to the stroke of the piston, plus the guide portion and clearance. This trunk is encompassed by a second, formed with or secured to the piston; and a third trunk of the largest size encompasses the piston trunk. It will be understood, therefore, that a *clearance* is preserved around the surfaces of the guide and piston trunk, and therefore steam can pass between them to operate on the ends, by which means the *total* area of the piston is exposed to the pressure of the steam. Secured to the piston are two rods above and below the trunk,—seen in the elevation,—each of which are connected to a T-piece, which latter is prolonged within the guide trunk. The connecting rod is attached to the inner, or guide portion of the T-piece, and thus the action of the piston is transmitted to the crank pin. The exhaust steam passage is prolonged to the centre of the cylinder connection, and thus one steam-pipe only is requisite. The illustration under notice is in connection with the condenser shown by Fig. 87, in page 260; and the proportions of the cylinder, trunks, &c., are now introduced:—

	Ft.	In.
Diameter of Cylinder	4	7
Length of Stroke of Piston . . .	2	6
Internal Width and Depth of Guide Trunk, $14\frac{3}{8} \times 16\frac{3}{4}$ ins.		
Width of Space between Trunks . .	0	$0\frac{3}{4}$
Length of Connecting Rod . . .	5	9
Depth of Piston	0	$5\frac{1}{2}$

For direct acting engines the single piston rod arrangement is the most efficient,—fully explained in pages 49, 50, and 51. The principal constructors of the type under notice are Messrs. Humphrys, Watt, Ravenhill, and a few firms in Scotland.

The elevation and plan shown by Fig. 117 is an ordinary design, being very similar to

Fig. 117,



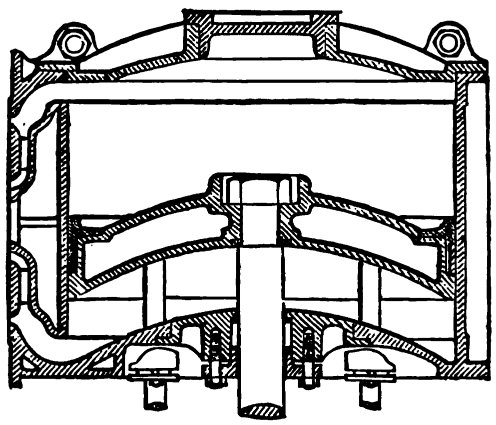
ORDINARY CYLINDER AND SINGLE ROD PISTON.

that fitted by Messrs. Ravenhill in H.M.S. "Enterprise," and not very unlike the practice of Messrs. Humphrys. The difference in the designs by the two firms alluded to consists, that Messrs. Humphrys recess the flange of the boring hole, or the stuffing box portion, internally, and the exhaust steam opening is at the front, instead of the top of the cylinder. Both, and indeed all the firms adopt double-gland stuffing-boxes for the piston rod, as shown in the illustration. A cylinder for a pair of engines 160 horse power nominal collectively, is of the following dimensions:—

	Ft.	In.
Diameter of Cylinder	3	9
Length of Stroke	1	6
Diameter of Piston Rod	0	6
Depth of Piston	0	6

The sectional plan of a cylinder, shown by Fig. 118, is an example by Messrs. Watt. In

Fig. 118.



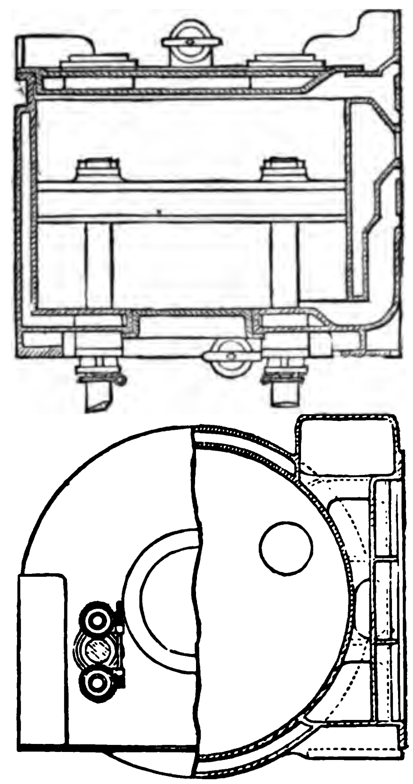
MESSRS. WATT'S CYLINDER AND SINGLE ROD PISTON.

this case, it will be seen, both the cover and piston are curved, to allow a given length for the connecting rod, and the cover of the boring hole is at the back end of the cylinder. The remainder of the design can be readily understood. For engines of 200 horse power nominal, each cylinder is of the following dimensions:—

	Ft.	In.
Diameter of Cylinder	4	2
Length of Stroke	2	0
Diameter of Piston Rod	0	6
Depth of Piston	0	10

Double piston rod cylinders for return-action engines are next to be noticed. An example by Messrs. Maudslay is illustrated by Fig. 119. The rods are above and below the centre line, sufficiently to clear the diameter of the crank shaft, and of such a distance transversely apart, as the width of the cranks and pin determine. The exhaust steam passage is on the top of the cylinder, and prolongec the front end. The body of the cylinder ε its ends are jacketed, similar to Messrs. Pen

Fig. 119.



MESSRS. MAUDSLAY'S CYLINDER AND DOUBLE ROD PIST

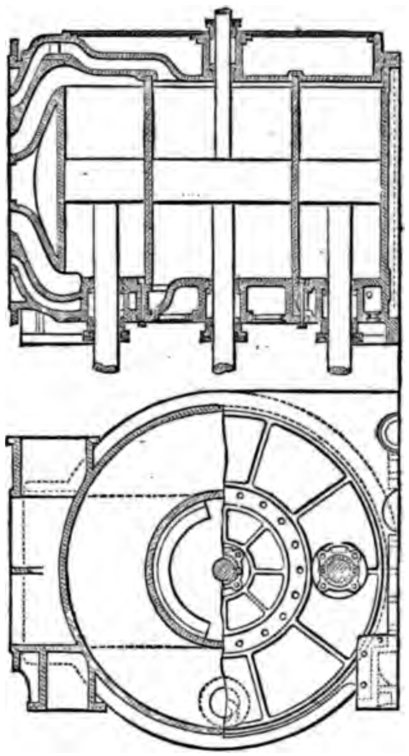
example, as seen by Fig. 115—page 274—ε the boring hole door is at the front end. 1 adjustment of the glands is attained by a no arrangement, seen in the elevation, and ill trated on a larger scale by Fig. 124, in pε 279. Messrs. Maudslay's practice for a cy der 450 horse power nominal is thus:—

	Ft. 1
Diameter of Cylinder	7
Length of Stroke	4
Diameter of Piston Rods	0
Depth of Piston	1

The compound system, or the adoption two cylinders in the place of one, for expans purposes, must next be alluded to. Amon

he firms who advocate the arrangement, Messrs. Randolph and Elder occupy a prominent position. Messrs. Dudgeon have lately put forth their knowledge on the subject, of

Fig. 120.



MESSRS. DUDGEON'S COMPOUND CYLINDER WITH THREE PISTON RODS.

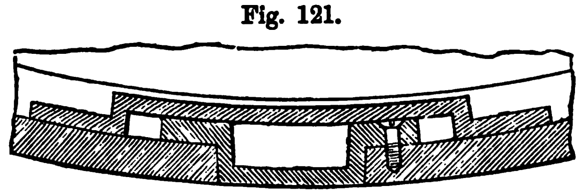
which the illustration, Fig. 120, is a good example. This is two cylinders, one within the other, and steam ports and passages to correspond, as shown in the sectional plan. Three piston rods are requisite, and an annular piston besides the one central. The piston rods are in the horizontal line—being for a direct acting engine—the glands are of the ordinary form, adjusted by nuts and bolts. The smaller cylinder is secured at the front end only, the back extremity being recessed in the back cover of the larger cylinder. Messrs. Dudgeon's practice for engines, on the compound system, 350 horse power nominal collectively, for twin screw propulsion with four cylinders, is as follows:—

	Ft.	In.
Diameter of Low Pressure Cylinder . .	5	2
Ditto High ditto ditto . .	2	3
Length of Stroke	2	0
Diameter of Low Pressure Piston Rods .	0	$4\frac{1}{8}$
Ditto High ditto ditto Rod .	0	$3\frac{7}{8}$
Depth of Piston	0	8

STEAM PISTONS.

The principles to be observed in connection with the details under notice are expressed in page 228, and an illustration of the gasket packing is shown by Fig. 59, in page 229. An example of a nearly similar mode of packing is shown by Fig. 121, being a sectional

Fig. 121.



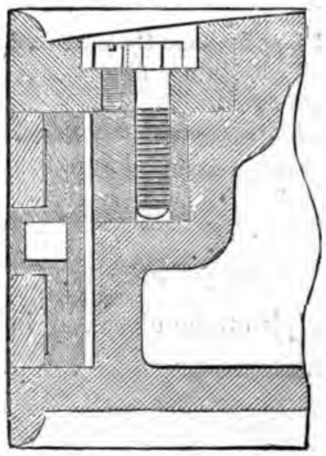
DUPLICATE MODE OF CONNECTING PISTON RING.

plan. The ring is divided, and the space filled by a recessed plate secured only at one extremity —to admit expansion and contraction. To prevent a collapse at the division, a second plate is introduced of a greater length, and thus the uniform curve is preserved. The transverse section, shown by Fig, 122, in page 278, illustrates the face ring, and the securing and stop studs, also the depths of the piston and ring.

The adoption of springs behind the piston ring is now becoming general, and more particularly where high pressure or super-heated steam is used. An illustration of the application of springs is represented by Fig. 123 for

a compound engine—being the pistons in connection with the cylinders depicted by

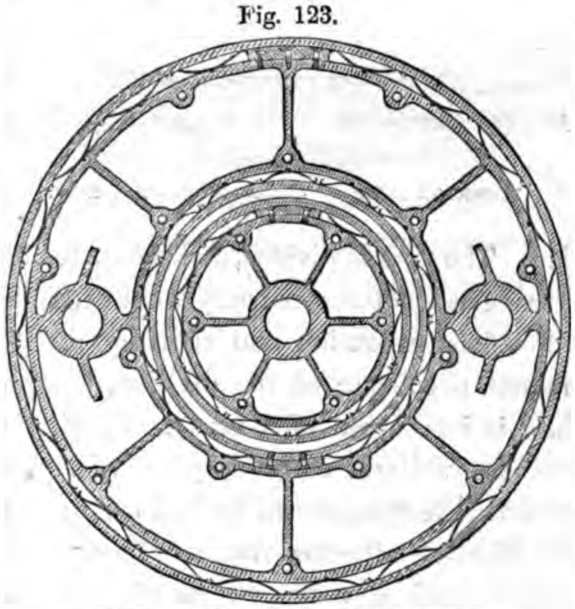
Fig. 122.



PISTON PACKING.

Fig. 120, in page 277. A sectional plan is only shown, as from that view the particulars

Fig. 123.



MESSRS. DUDGEON'S COMPOUND PISTON.

are best rendered. The springs, it is seen, act against each ring, equally, on each side of the division, suitable connecting portions being provided to prevent the passage of the steam. It may be added that the ordinary face ring with the requisite studs complete the arrangement.

Steam used as a spring is obviously certain, when it is remembered that the lead of the slide valve causes an admission of steam into the cylinder, in advance of the piston, and thus prevents a concussion, with quick velocities. It is from this effect that some engineers have advocated the use of steam behind rings, and thus dispense with the packing and metallic springs.

Mr. Miller, in the year 1862, read a paper on this subject to the members of the Institution of Mechanical Engineers. His idea of perfection was the adoption of "Mr. Ramsbottom's" rings with steam acting behind them, instead of trusting to the elasticity of the steel. Now this mode may be applicable with pistons of small diameter, but for cylinders 100 or 150 inches diameter, the unequal action of the steam, with unequal effect, will cause a leakage more or less as the imperfection exists. Further than this, it must not be forgotten that the exhaustion of the steam will cause the rings to contract, although opposite passages may be provided in the piston for the opposite action of the steam. What is required with a piston is, a constant contact with the internal surface of the cylinder; and steam from within the cylinder cannot effect this without valves are introduced in the piston, and even then, the liability of their disarrangement tends to doubt their utility.

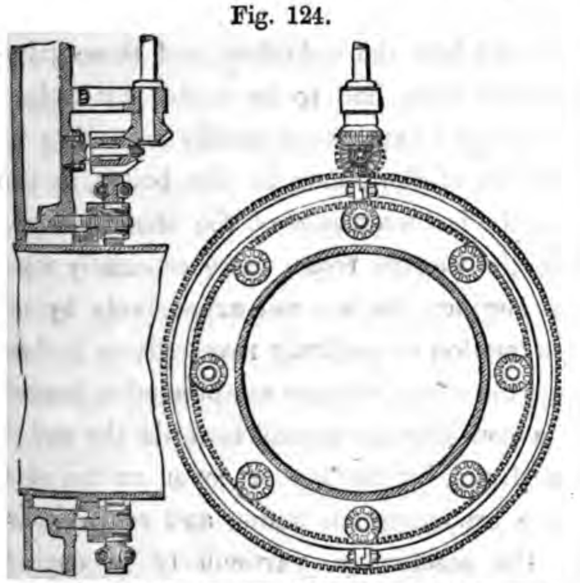
Referring again to Mr. Ramsbottom's pistons, that gentleman has moved a step in the right direction; but while admitting it, there is no cause for wonder at the result, for his invention is simply the division of power

and surface to produce a perfect contact for the whole. Now with a deep ring, or a single surface in contact, a proportionate amount of power is requisite to keep the ring to its work; but when the surface is subdivided—or a series of shallow rings—the effect is more practically produced, simply because the power is divided also. This class of piston has met with much favour for locomotives, and is becoming more in use for marine purposes than was once anticipated.

MECHANICAL MEANS FOR THE ADJUSTMENT OF THE GLANDS OF STUFFING BOXES.

When the packing around the trunk of an engine requires to be tightened, the gland must be shifted; and to do this effectually each nut should be simultaneously turned. Now with separate manipulation—the general practice—this is a tedious as well as a dangerous

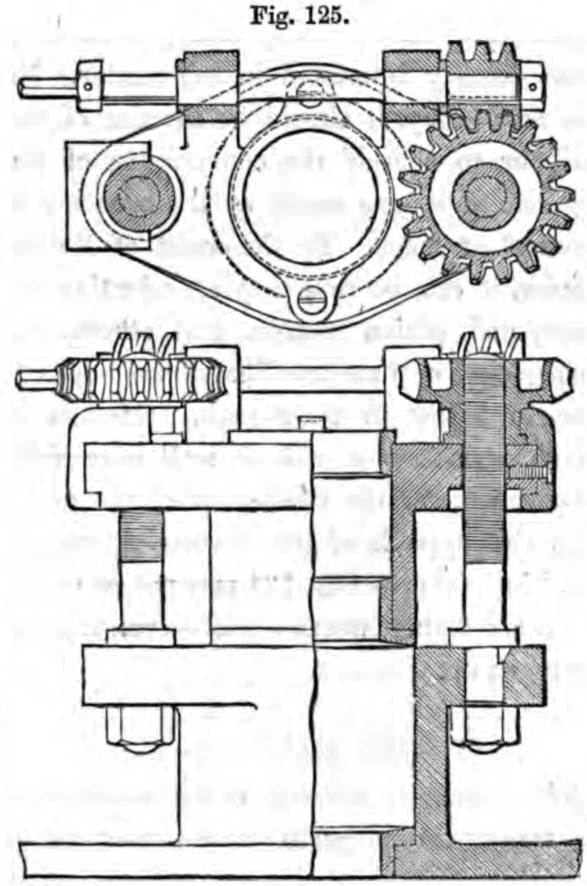
Fig. 124.



BURGH'S PINION AND WHEEL MOTION FOR ADJUSTING TRUNK GLANDS.

operation when the engine is in motion. To obviate these evils, we have designed the arrangement of pinion and wheel motion as shown by Fig. 124. This is a double-toothed ring, the inner teeth being in gear with pinions, which are virtually nuts screwed on as many studs. The outside teeth of the ring are in gear with a spur pinion, and this latter is mitred at its extremity, a second mitre pinion imparting motion when required. The action is therefore as follows—the perpendicular rod is prolonged to the top of the platform or cylinder, surmounted with a cranked handle; on imparting motion to the mitred pinions,

Fig. 125.



WORM AND PINION MOTION FOR ADJUSTING PISTON ROD GLANDS.

the spur pinion causes the ring to revolve, when each screw pinion acts against the

gland, thus compressing the packing equally throughout its entire circumference.

A similar effect is attained for the packing of piston-rod stuffing boxes by the arrangement represented by Fig. 125, in page 279. This is a worm and pinions, and on motion being imparted to the former the latter revolves, and thus shifts the gland in the required direction. This arrangement was first introduced by Messrs. Maudslay, and they have adopted it with much success.

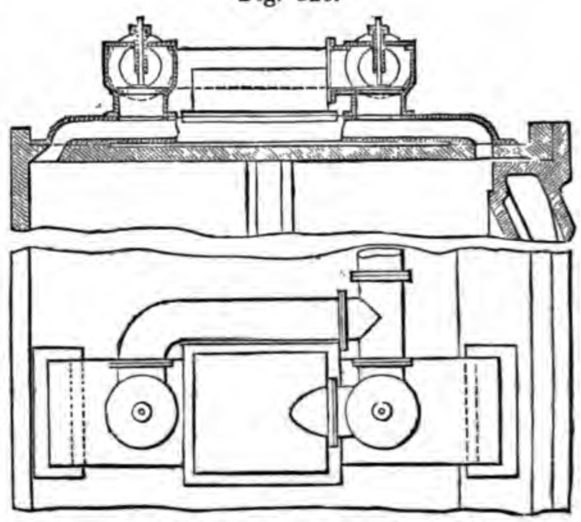
The benefit of being able to tighten the packing when under steam, or the engine in motion, can doubtless only be appreciated by those directly interested in the matter. Be this as it may, it should be the aim of the designer to consult the convenience of the engineer at sea, as much as the economy of material at home. To the credit of Messrs. Napier, it can be said they are adopting the worm and pinion motion, and others will perhaps in due time sacrifice a little expenditure to follow in their path. Makers of trunk engines, also, will do well to consider this matter, for the tightening of the gland with that type is of the utmost importance. The hint given by Fig. 124 may not be novel, but it is certainly worth a trial—even if to be improved on.

CYLINDER RELIEF VALVES.

The principles relating to the details now under notice cannot perhaps be better expressed than in Mr. Penn's specification of his "Patent Relief Valves," of which Fig. 126 is an illustration. After the legal phraseology, Mr. Penn proceeds as follows:—

"This invention has for its object improvements in escape or relief valves to the cylinders of marine and other steam engines. Escape or relief valves are now commonly fitted to the cylinders of marine and other

Fig. 126.



MESSRS. PENN'S PATENT CYLINDER RELIEF VALVES.

engines, to allow of the escape of water should any be carried over from the boilers or otherwise get into the cylinders, and these valves hitherto have had to be loaded by weights or springs to an extent greatly exceeding the pressure of the steam in the boiler, to prevent the inconvenience of the steam escaping into the engine room, more especially when the engines are worked expansively by the link motion or suddenly reversed; as in these cases the steam becomes compressed or jammed to a considerable extent towards the end of the stroke, by the lap or cover on the slide valve producing an undue and severe strain on the machinery, particularly in engines where the slide valves are prevented coming off their faces by being fitted with packing on the back to relieve the pressure on the faces.

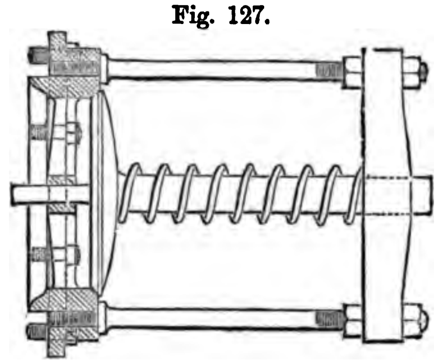
"According to my invention, I apply, instead of weights and springs, the pressure of the steam to keep the escape or relief valves in their seats. For this purpose I place the valves, enclosed in cases, on any convenient part of the cylinders sufficiently high for any water to pass by means of pipes into a separator in the main steam pipe of the engine, or to the main steam pipe itself, where the position of the latter will admit of the arrangement. It will be seen that the valves are kept in their seats by the difference between the pressure of steam in the main steam pipe and the cylinder, and also in consequence of the difference of area between the outer and inner surfaces of the valve.

"In this arrangement there is so little weight tending to keep the valve on its seat, that it will frequently open in consequence of the compression of steam in working expansively by the link motion, or arising from suddenly reversing the engines, and no strain can come on to any part of the machinery beyond that resulting from the pressure of steam in the boiler.

"The ordinary relief or escape valves, on the other hand, are often rendered inoperative by being overloaded to prevent the steam escaping into the engine room, and this is the cause of their frequently sticking fast. It will be seen that by my arrangement the objection of working engines expansively by the link motion, and which now causes the steam to become much compressed at each end of the stroke, is removed, the steam as soon as the compression commences passin back into the steam pipe without any loss. All accidents from scalding will also be avoided."

The ordinary spring relief valve adopted by Messrs. Penn is represented by Fig. 127.

Fig. 127.

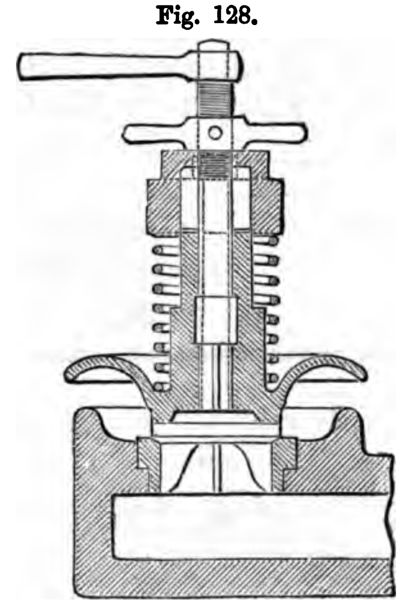


MESSRS. PENN'S CYLINDER RELIEF VALVE.

The valve is the disc type with a spring at its back, and two side rods sustain the guiding or cross bar. The seating is shown, in section, and also the inner guide for the smaller spindle.

The practice of Messrs. Maudslay being always worthy of notice, therefore their view of

Fig. 128.



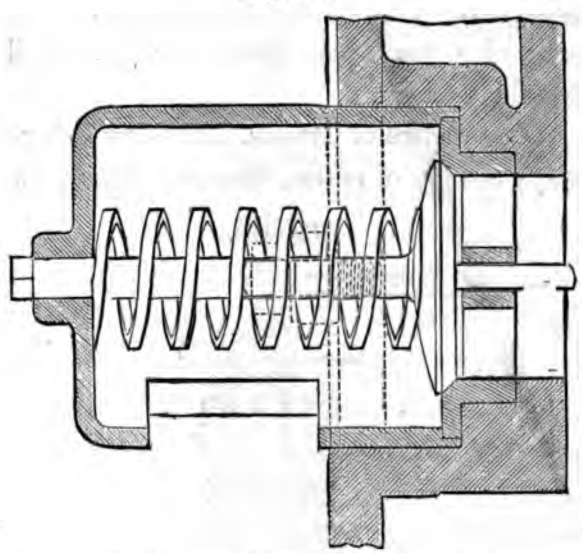
MESSRS. MAUDSLAY'S CYLINDER RELIEF VALVE.

the present subject is represented by Fig. 128. The valve is guided below the seat by three ribs, and the spindle above is inserted in

the guard or splash disc. This latter portion is to prevent the condensed steam from scalding the attendants, and scattering itself in the engine room. The means of adjusting the spring is by a telescopic cross piece, above the spring, and the set handle retains the required position for both. The handle at the extremity of the valve spindle is to prevent the valve from turning during the adjustment of the spring.

The constant means of adjusting the spring is not always essential, as the example, Fig. 129, by Messrs. Ravenhill, indicates. This

Fig. 129.



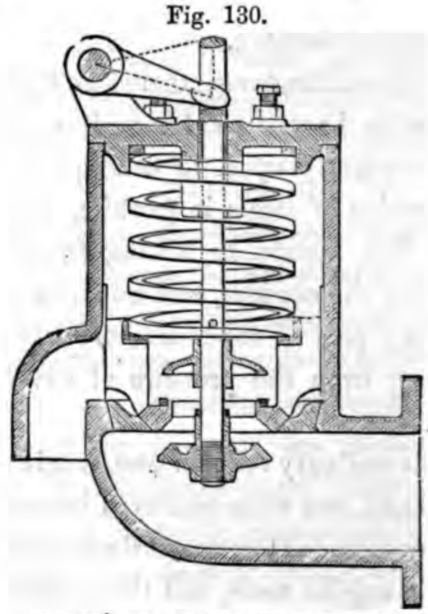
MESSRS. RAVENHILL'S CYLINDER RELIEF VALVE.

is an ordinary valve and springs enclosed in a casing, and the securing of the latter alone compresses the spring. The advantages with the casing are, that the spray, when the valve is opened, is transmitted direct into the bilge; and the spindle and spring being enclosed within the casing, are thus guarded from fractures by the fall of any weighty body.

Independently of the valves under notice, relief plug valves are often attached to the cylinders, their use being that, in the event of blowing through, or extreme priming from the boilers, a simultaneous discharge can be effected. Messrs. Humphrys introduce an addition to these valves beyond the plug, which is an india-rubber flap valve, the closing of which excludes the air from the cylinder; thus the vacuum is unaffected while the plug valve is opened.

Some consideration to this matter has been bestowed by ourselves, and the result is the production of the next two examples of relief

Fig. 130.

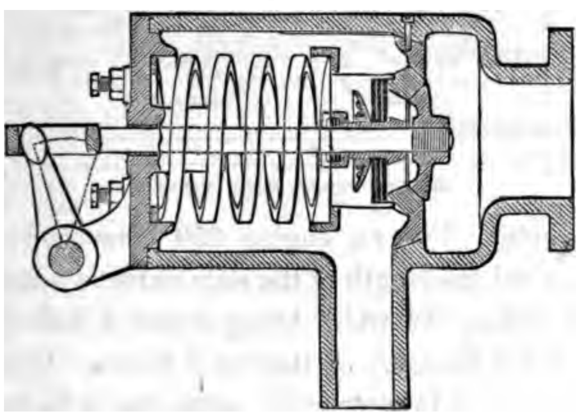


BURGH'S CYLINDER RELIEF VALVE.

valves. The illustration, Fig. 130, is an elevation of a triple disc valve, which embraces all the advantages of the spring and plug valves, while dispensing with the latter. The positions of two of the valves, as shown, are as required when blowing through. On the return stroke of the piston, the upper or return vave will fall on the india-rubber seatings, and thus the

vaccum will be unimpaired. When the spring is required to act, the lower or spindle valve is closed, and the annular or spring valve is as effective as an ordinary solid disc valve. The lever on the cover of the casing is manipulated at the starting platform, and the slot in the valve spindle ensures a certain action. To render this arrangement further obvious, the illustration, Fig. 131, is introduced.

Fig. 131.



BURGH'S CYLINDER RELIEF VALVE.

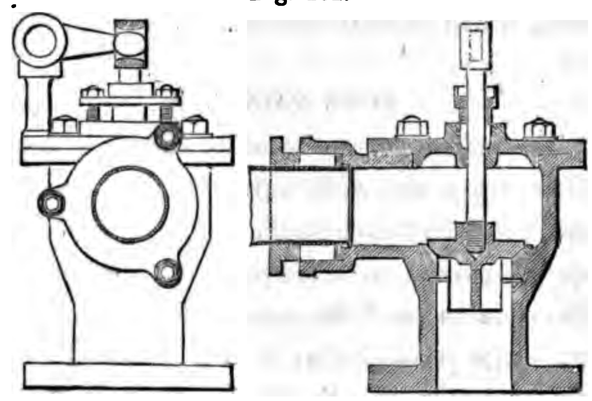
This example shows the spindle valve closed, and in the place of a metal return valve, an india-rubber disc is adopted, with an ordinary guard, and a stuffing box to prevent a leakage around the spindle. The springs in both of the examples can be adjusted by the set studs on the cover.

BLOW-THROUGH VALVES.

To warm the cylinders and cause a vacuum in the condensers before starting the engines, supplementary valves are requisite. An example for this purpose is illustrated by Fig. 132. This is an ordinary disc valve with a spindle and lever, and the casing of a general form completes the apparatus.

It is obvious, on reflection, that steam, for the purpose alluded to, should be admitted on

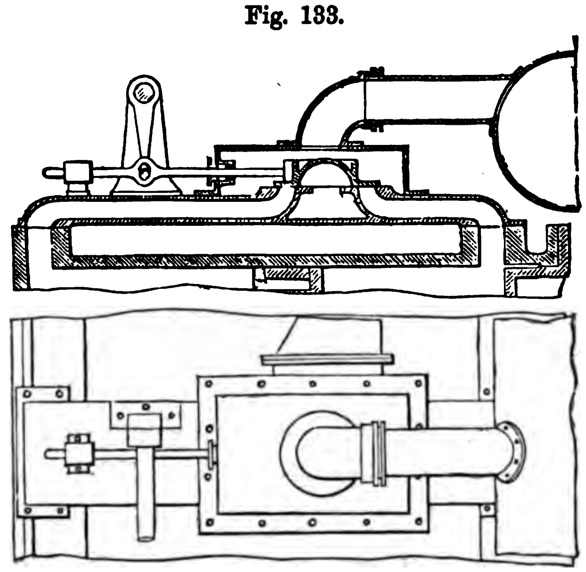
Fig. 132.



ORDINARY CYLINDER BLOW-THROUGH VALVE.

each side of the piston, after which into the condenser, and to attain this a corresponding number of valves or pipes are requisite. To suit this requisition Messrs. Penn have adopted the valve and passage seating shown by Fig. 133.

Fig. 133.



MESSRS. PENN'S CYLINDER BLOW-THROUGH VALVE.

As the admission and exhaustion of the steam into and from the cylinder is the main effect, an ordinary slide valve is preferred, by which adoption a simple arrangement results. It

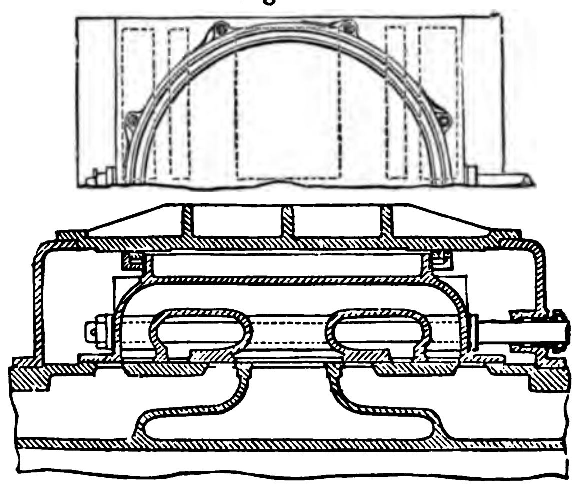
will be noticed, however, that the steam cannot be admitted at both ends of the cylinder simultaneously. This, however, is no great fault, which practice has proved.

SLIDE VALVES.

The steam engine becomes, indeed, a poor affair when the slide valve is out of order, and it is from this cause that many productions are introduced to accomplish the best effect. The slide valve of the present day has grown into large proportions; it is a common practice to make the length five feet, and the width six feet, for marine purposes. Now, with these dimensions, obviously a large surface is exposed to the action of the steam. The arrangement and formation of the ports, passages, and bars, are simple matters compared with the production of a perfect and constant working contact for the respective surfaces. To accomplish this, the action of the steam must be duly considered, and the mechanical means for adjustment.

The valve mostly used in the present day is the type known as the double-ported equilibrium. An example of this kind is shown by Fig. 134, as arranged by Messrs. Penn. The valve is depicted at half stroke, so that the ports are covered. When the valve is at the full stroke, the width of the opening caused equals half of the supply port. The steam admitted to the inner ports passes through the valves, and thus the pressure is relieved from the surfaces in contact. To prevent the steam acting on the back of the valve, a packing ring is located thereon—shown in the section and in the half plan. The valve rod passes through the body, and is secured at the extremity by a nut. The casing is the ordinary design, and secured by studs and nuts to the

Fig. 134.

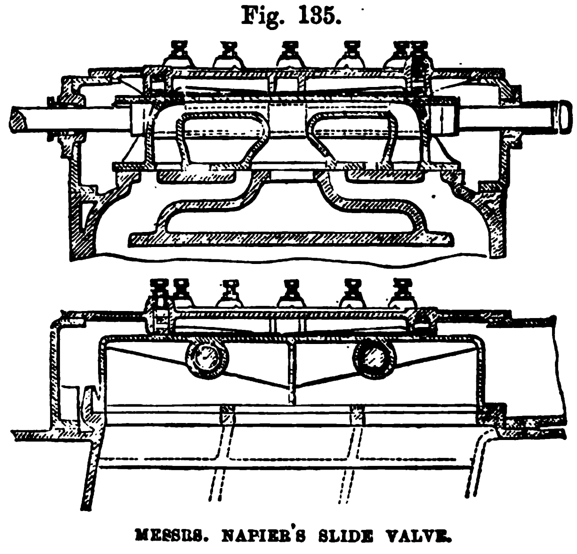


MESSRS. PENN'S SLIDE VALVE.

cylinder. For an engine 250 horse power nominal, the length of the slide valve is 5 feet $4\frac{5}{8}$ inches, the width being 4 feet 2 inches, and the diameter of the rod 3 inches. This example is in connection with the cylinder depicted by Fig. 115 in page 274.

Messrs. Napier's practice for the present details is represented by Fig. 135, both in

Fig. 135.



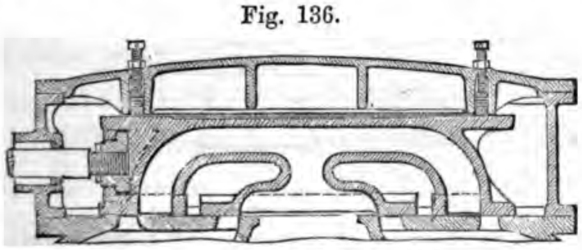
MESSRS. NAPIER'S SLIDE VALVE.

"longitudinal and transverse sections." This

firm, in common with other engineers, use two rods, not only to transmit the motion, but also to retain a direct action. The rods pass through the valve, and are prolonged through the back end of the casing. A bonnet or cap encloses the rods, and thus stuffing boxes are obviated. The valve is further guided by a flange cast with the cylinder on the lower side of the facing, and a projection on the upper. The valve thus works between guides throughout its motion, independently of the support derived from the prolongation of the rods. In the longitudinal section the valve is shown as admitting the steam at the front end of the cylinder. It is almost needless to state this example is the double-ported equilibrium type. The firm under notice have fitted this example into three armour-cased frigates for the Turkish Government, named the "Osman Ghazy," "Abdul Aziz," and "Orkan." The length of this valve is 5 feet $2\frac{1}{4}$ inches, and the width between the guides 5 feet $9\frac{1}{2}$ inches, the diameter of the rods being 4 inches.

In connection with the surface condensers shown by Fig. 92—page 264—is the slide valve of one of the cylinders now represented by Fig. 136. Messrs. Rennie have designed

Fig. 136.



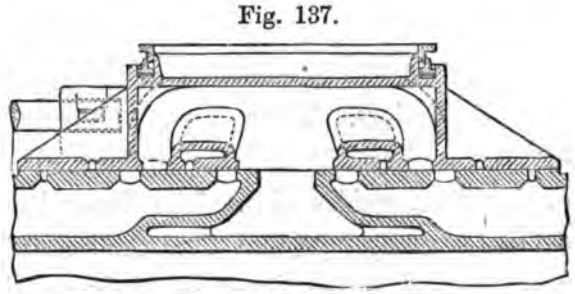
MESSRS. RENNIE'S SLIDE VALVE.

a practically correct form for the exhaust steam passages, with the cylinder ports for supply and exhaust. The mode of connecting the rod is by a nut recessed in the front part of the valve, centrally of its width, the rod being further sustained by a stop or check nut. The cover of the casing, similar to the previous example, is strengthened internally, and thus an even surface for the exterior is preserved—the adjusting and securing studs and nuts excepted. The length of the valve is 3 feet 6 inches, the diameter of the screwed portion of the rod $2\frac{1}{4}$ inches, and the remainder $2\frac{7}{8}$ inches, one rod being adopted.

Messrs. Maudslay, Watt, Ravenhill, Humphrys, and other firms, adopt the type of valves under notice; each separate production being alike in the principle of its action, if not in the design.

The half travel of an engine cylinder slide valve equals, of course, the outside lap + the width of the supply steam opening. Now, as this fact is unalterable, the subdivision of the ports in question must reduce the total travel of the valve. Entirely ignoring, therefore, the too common belief that the steam must be wire drawn by this subdivision, the Messrs. Maudslay have lately introduced the three-ported

Fig. 137.



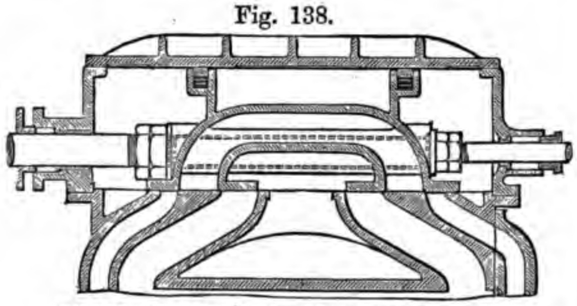
MESSRS. MAUDSLAY'S THREE-PORTED SLIDE VALVE.

valve, illustrated by Fig. 137, with great success. It will be seen that there are two large bars in the cylinder, and three in the valve on

each side of the central or exhaust ports. Corresponding with this, three ports in the valve are formed—two for the supply, and one for the exhaust steam—and the ports in the cylinder are for similar purposes. The supply steam passages in the valve are contracted at the centre, similar to the example shown by Fig. 135, by which form the area of the exhaust passage is enlarged in proportion to the depth of the valve. The length of a three-ported valve for a cylinder equal to 50 horse power nominal, is 3 feet 8 inches, and the width about 3 feet 10 inches, the diameter of the valve rod being $2\frac{3}{8}$ inches.

In page 277 the cylinder for a compound engine is represented by Fig. 120, and in connection with it the slide valve is now illustrated by Fig. 138. This detail is a double

Fig. 138.

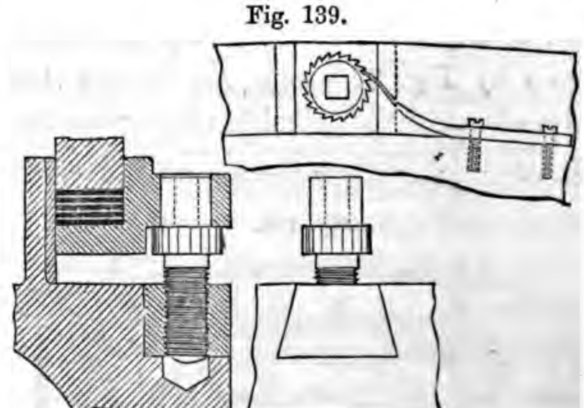


MESSRS. DUDGEON'S COMPOUND SLIDE VALVE.

single-ported valve, the passage within the body being for the interchange of the steam from the cylinders. The rod passes through the valves and casing, and the stuffing boxes are, therefore, guides also. The inner bars have scarcely any laps, and thus the "time" for supply and exhaustion are nearly equal. The position of the valve on the rod—to alter the lead at either end—is adjusted by nuts, and check nuts are added to prevent looseness.

The power requisite to move a slide valve can be readily known by the most simple formula—being a matter of ordinary multiplication and addition; and this is apparent on remembering the principles of the case. For suppose it be required to know the force to overcome the inertia of a given load: the value of the load is the first step, and this is merely a matter of surface × pressure + gravity; next, the coefficient of the friction must be considered, and the result is the power required. Then as the pressure and surface form the largest sums, the reduction of the latter exposed to the direct action of the former is the only means of lessening the power. Now, engineers have not long discovered, or rather, put into practice, the fact that the steam which can cause the valve to adhere to the cylinder facing can also be used to obviate that adhesion; and thus in each of the examples of slide valves illustrated, packing at the back is introduced to prevent the steam from acting thereon. Considering first Messrs. Penn's practice, an example of their mode is

Fig. 139.

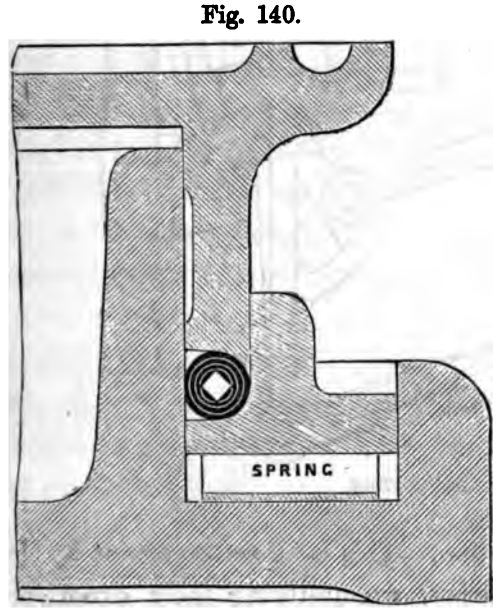


MESSRS. PENN'S MODE OF ADJUSTING THE PACKING RINGS FOR SLIDE VALVES.

depicted again in detail by Fig. 139. The face ring surrounds the projection on the

valve, and is packed with india-rubber contained in the packing ring. This latter ring is adjusted by set screws, and their looseness is prevented by ratchets and springs, which also indicate—by the clicks—the movement of the screw. Messrs. Napier's and Rennie's practice is shown by Figs. 135 and 136, in pages 284 and 285. Both firms, it will be noticed, prefer the set screws outside the casing, and the face ring is recessed in the cover. Messrs. Napier adopt spiral springs around each screw, and two packing rings with the packing between them. Excepting the springs, Messrs. Rennie's mode is precisely similar. In page 285, Fig. 137 is alluded to as Messrs. Maudslay's valve. They use a novel packing ring, shown at a large scale, in section, by Fig. 140. This is a face ring of

Fig. 140.



MESSRS. MAUDSLAY'S PACKING RING FOR SLIDE VALVE.

peculiar section packed with india-rubber packing, and the ring enclosing the same is supported on curved springs of steel. This arrangement is self-adjusting, and the firm have adopted it for some time without failure in any case. It will be seen, from these examples, that the back of the valve is unexposed to the steam; and to further relieve the face surface, a communication with the condenser and the annular space within the packing ring is sometimes introduced. By enlarging the face ring the steam can be arranged to lift the valve, if desired, from the facing, and thus reverse the situation of the friction. Another example of self-adjusting packing of a more simple form than Messrs. Maudslay's is shown by Fig. 138, in page 286. Besides the Messrs. Dudgeon, Messrs. Penn have often used it, being simply a ring of metal packed with india-rubber under it.

SLIDE VALVE LINK MOTION.

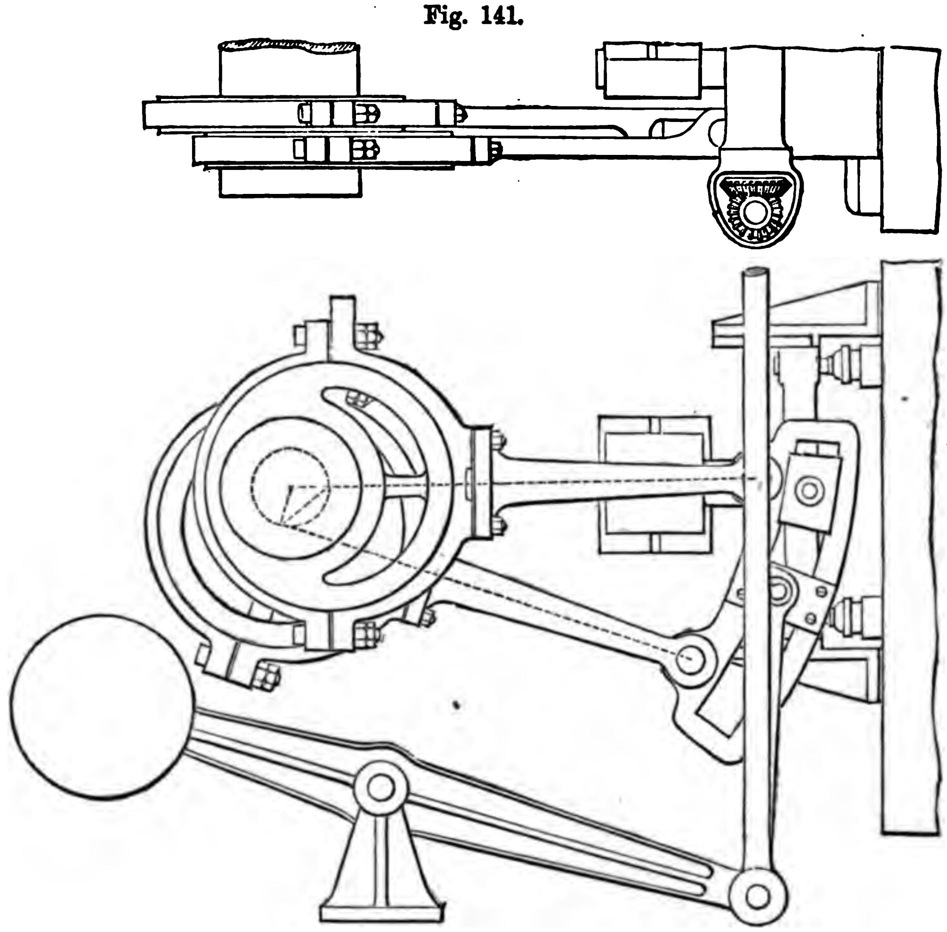
It is, of course, well understood that the utility of the portion of the marine engine now to be noticed is a combination of detail to produce three effects at will—starting, stopping, and reversing. Now, were it only requisite to produce the means for starting and stopping, the mechanical appliances would be a simple matter; but when reversing must be accomplished by the same gear, the simplicity and effectiveness of the arrangements demand closer attention. In common with the other claims on marine engineers, they have not been behind in producing what was requisite in the present case; each maker escaping each other in design, but yet all accomplishing similar results.

The first example now brought into notice is that by Messrs. Penn, lately fitted by them in H.M.S.S. "Northumberland," represented

by Fig. 141. The link is the "slotted" kind, the eccentric rods being connected within the extremities, and the suspending point is at the centre of the link. The eccentrics are the usual type with gun-metal bands and wrought iron rods. The link is raised and lowered by a screwed rod receiving motion from mitre gearing on the hand wheel shaft. The weight of the link, pins, rods, and sliding block is counterbalanced by the lever and balance weight below the motion; and thus the power requisite for shifting the "slide valve" is greatly reduced. It will be noticed that two slide rods are adopted, and the requisite guidance is above, below, and central of the rods, by which positions a direct action is certain. When the firm in question use a single rod for the valve, as shown by Fig. 134—page 284—the central guide is adopted only.

To be noticed next, also, as an improved example, the illustration, Fig. 142, is worthy of attention. This is an arrangement by Messrs. Maudslay, exemplifying their experience in the matter. It will be noticed that the principle of the action is nearly as that of Messrs. Penn's arrangement, but the design varies in many points. The lifting rod is connected to the centre of the link, and the rod is looped

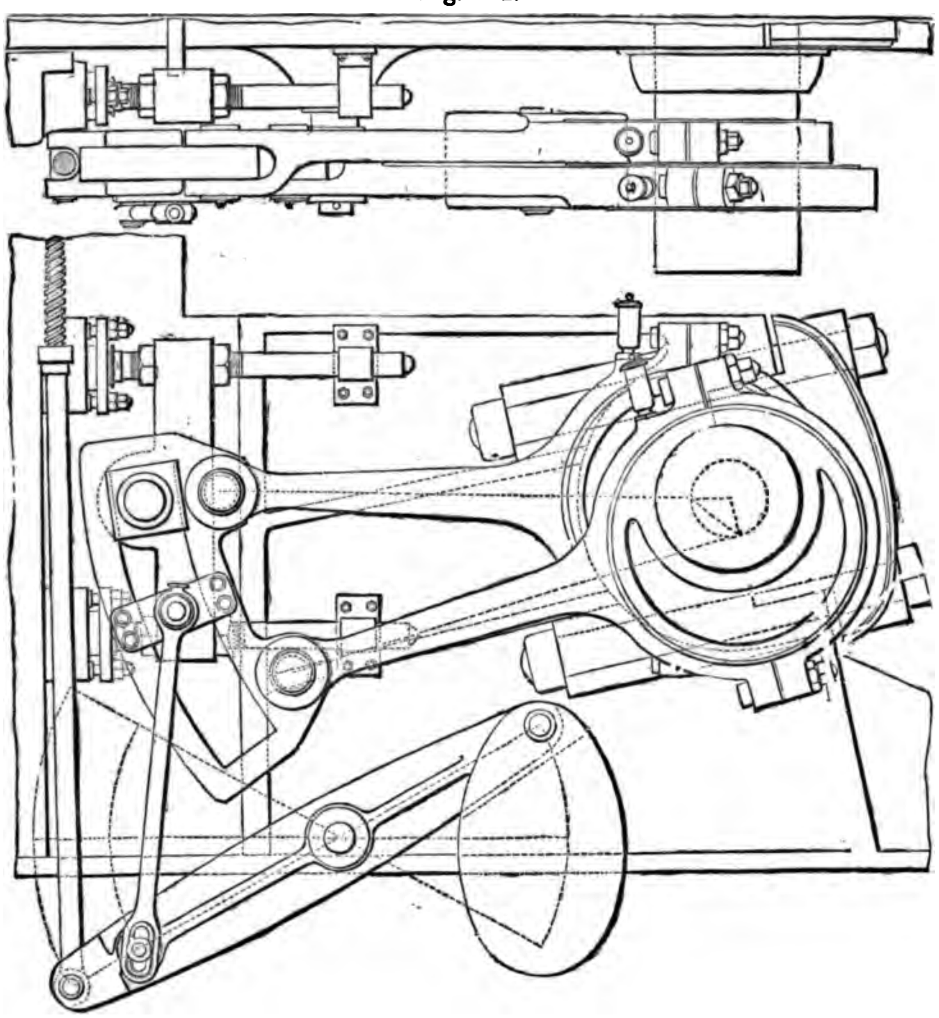
Fig. 141.



MESSRS. PENN'S SLIDE VALVE LINK MOTION.

at the connection with the lever, and thus the link rests on the sliding block. This is contrary to the connection in Fig. 141, where the rod is connected alike at each extremity, and the link hangs on the connecting pin. The position of the screwed rod in Fig. 142 is behind the link, and thus the lifting rod is in advance, or two separate connections with the balance lever. Messrs. Penn, it will be seen, prefer a single connection, or at the extremity of the lever. The slide rods are guided by brackets secured to the main frame—the latter being shown in the plan and elevation. The central guide for the cross bar, seen in Fig. 141, is omitted in this instance; and the block pin is fixed behind the bar, rather than nearly centrally of its width, as adopted by Messrs. Penn. Messrs. Maudslay prefer the eccentric rod and half band to be in one forging, lined with gun metal where in working contact. The counterbalance is hung on the lever, rather than fixed, and the weigh-

Fig. 142.



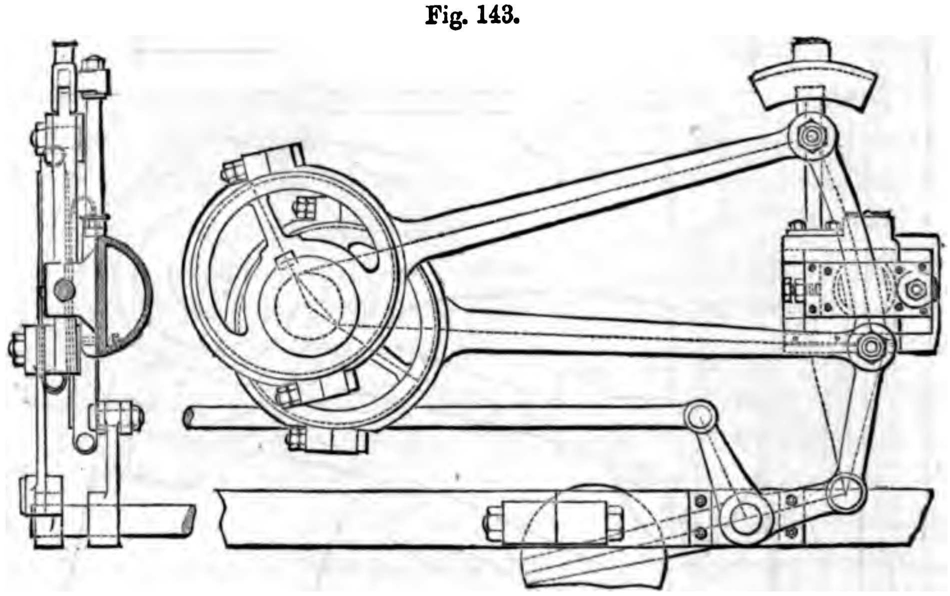
MESSRS. MAUDSLAY'S SLIDE VALVE LINK MOTION.

shaft is supported in the main frame, thus dispensing with the bracket used by Messrs. Penn. Excepting these differences of opinion alluded to, both the firms have agreed conclusively on the same subject.

Amongst engineers who have studied the present subject, the late Mr. E. Humphrys claims notice; for that gentlemen preferred the arrangement shown by Fig. 143. In the place of the ordinary slotted link a solid type is preferred, the eccentric rods being connected to the front of the casing. The adjustment of the surfaces in working contact is ensured by bolts and nuts securing the loose or lap portions of the guide.

The guiding portions having been thus far explained, attention is next directed to the link and its connections. The eccentric rods are attached in the ordinary manner by being forked and clasping the link—bolts and nuts being used in the place of the usual pins and washers. The upper extremity of the link is

Fig. 143.



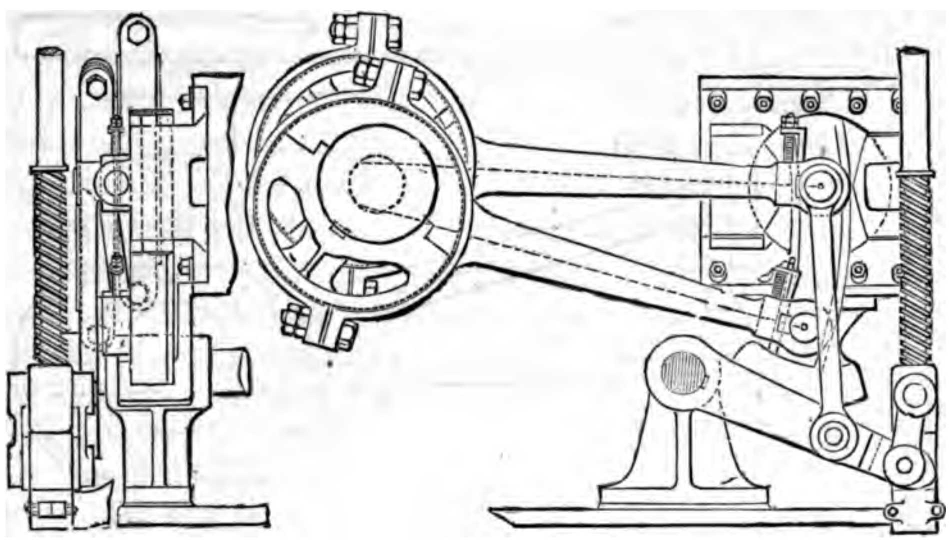
MESSRS. HUMPHRYS' SLIDE VALVE LINK MOTION.

at the extremities. The link passes through the block, the latter being in halves, curved at the outsides to permit the required oscillation. The adjustment of the wearing surfaces is accomplished by the set stud on one side, and the slide valve rod at the other; the rod being screwed in its connection with the portion enclosing the blocks—the nut outside the plate preventing looseness of the connection. The sliding portion is guided in a flat dovetailed guide, which latter is a hollow bracket secured prolonged beyond the rod's connection for the purpose of being guided when in the position shown. The guide is a curved slotted block supported on a standard or rod of wrought iron secured on the hollow bracket. The lower extremity is connected to the lifting rod, and the latter to the lever requisite to impart the required motion to shift the link; the other lever seen being in connection with the starting gear. The remainder of the details are so well understood from the drawing, that

further description will be more tedious than useful.

The firm to be alluded to next, as constructors of "solid" link motion, are Messrs. Ravenhill. Their arrangement will be understood by referring to the example, illustrated by Fig. 144, together with the following description. The link is dovetailed on one side, and passes through the block, which is in separate portions. The sliding piece sustaining the block is dovetailed at the back, and fits in a guide is unlike any of the previous examples, although almost similar details are employed. In this instance the screwed portion of the rod is level with the link rather than above it. The lever is connected by two short rods to a sliding block on the screwed rod—the lifting rod being connected to the upper eccentric rod's pin. Now, on certain motion being imparted to the screwed rod by the overhead mitre gearing, the screw block will rise on the screw, and, by its connection with the

Fig. 144.



MESSRS. RAVENHILL'S SLIDE VALVE LINK MOTION.

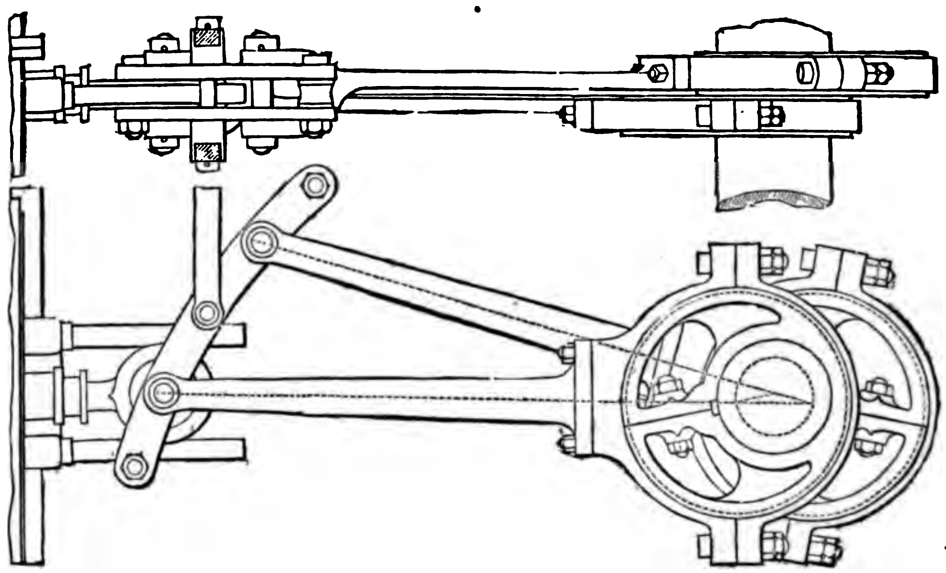
bracket of corresponding form; the adjustment of the guiding surface being ensured by the studs and nuts securing the plates. The centre line of the eccentrics, in plan, is a continuation of that of the slide valve rod; but the connection of the rods is at the side of the link, rather than clasping the same, as in the previous examples. The pins are fitted with brasses, and their adjustment is effected by keys and screws fitted on the rods. The arrangement for raising and lowering the link lever, raise the link. It will be remembered that Messrs. Humphrys guide the upper extremity of the link. Now Messrs. Ravenhill, it will be seen, guide the lower end by the bracket secured below the valve rod guide bracket.

The arrangement of link motion next to be noticed was originated in Scotland, and has gained favour enough to be atdoped by Messrs. Dudgeon, in London, in conjunction with the engines fitted in the M. S. "Ruahine." This

example is illustrated by Fig. 145, and the following description will render its advantages conclusive. The link is in two separate portions, connected at the extremities by bolts and nuts. The block is recessed on each side to receive the links, and by its enclosure in the valve rod a certain connection results. The upper and lower surfaces of the eye of the rod are guided by two rods secured above and below, keyed into bosses formed on the casing, and thus deflection is obviated. The eccentric rods are attached to the outer surfaces of the links, separate pins being requisite for each eye. The suspending or lifting rod is connected centrally of the link, and thus an uniform action is certain, irrespective of the direction of the crank's movement. With this arrangement there is certain advantages worthy of commens. The centres of the block and eccentric rod's pin are on the same point; also the link is suspended centrally of its length, and strength of material is gained by the peculiar form of the link.

It will be remembered that with all the previous examples alluded to, the link is directly in front of the casing, or on the same side of the crank shaft as the cylinders. This situation is, however, by no means compulsory, for Messrs. Napier and Rennie have often fixed the guide and suspended the link at the side of the condenser. The slide valve rod is prolonged to connect with the sliding piece supporting the block pin, and thus a return action results. This arrangement will be better understood by alluding to Fig. 146. The valve rod, it is seen, is above the crank shaft—the Messrs. Rennie prefer two rods above and below the shaft—and inside the eccentric rods; the guide piece is a flat plate, bevelled at the edges, working in a corresponding block secured to the side of the condenser. The link is the slotted kind, hung at the centre of its

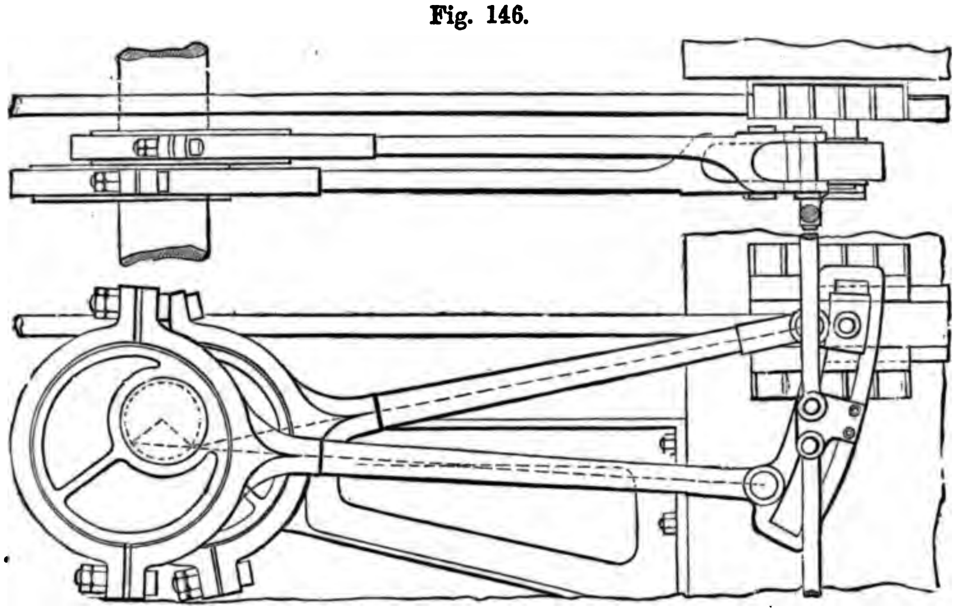
Fig. 145.



MESSRS. DUDGEON'S SLIDE VALVE LINK MOTION.

length by the lifting rod, the other extremity of which is attached to a sliding screw block above the link. The rod in the reverse direction is connected to a lever below, the shaft of which is in connection with the counterbalances and steam starting gear. This position for the link is particularly adapted for return action engines, where the position of the cylinders is as close to the cranks as the half stroke of piston and clearance will permit. It is also certain in its application when *long* eccentric rods are required.

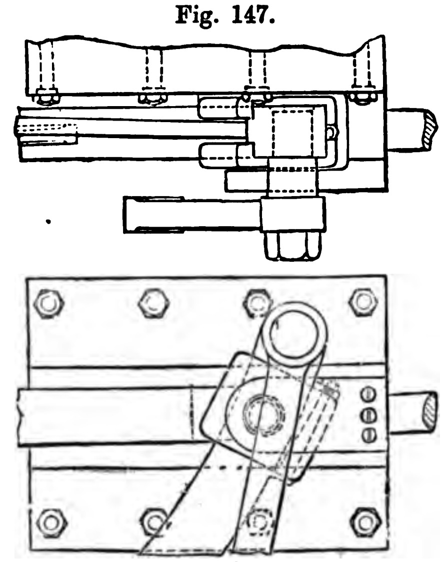
Fig. 146.



MESSRS. NAPIER'S SLIDE VALVE LINK MOTION.

The connection of the valve rod is an important item in the arrangement of the motion under notice, and it will be observed that only Messrs. Humphrys and Dudgeon adopt a central connection for the link and slide valve rod. By Fig. 147 a mode of connecting the solid link and the valve rod is represented, as adopted by the author. The block encloses the link, and has projections on each side for support, each of which are inserted in the forked portion of the valve rod. The block is open at the front to allow the centre of the eccentric rod to be on the centre of the block both in elevation and plan. The adjustment of the wearing surfaces of the block is insured by a back plate and key, the front surfaces being steeled to cause equal wear at that

Fig. 147.

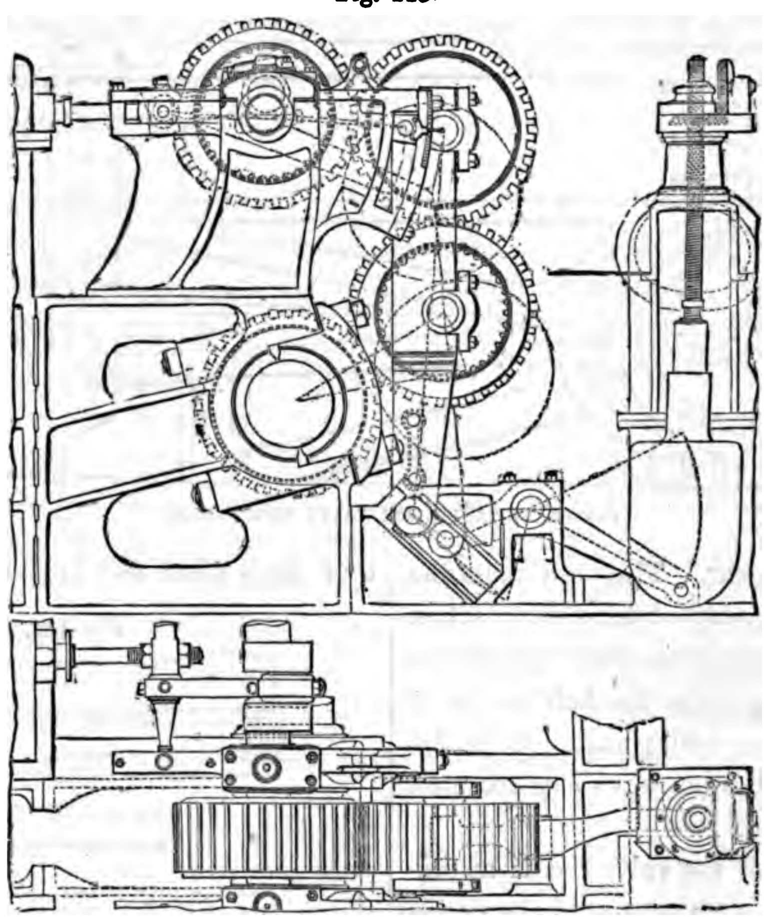


BURGH'S MODE OF CONNECTING THE LINK AND VALVE ROD.

part. The link is slotted at the extremities to receive the rods, but solid between; thus a single eye eccentric rod is available, combined with a single solid link and central connection.

Common to all engineers is the fact that the cause for raising and lowering the link is to affect the position of the slide valve. Now to reverse the principle of this, Mr. C. Sells has patented an arrangement of spur wheel motion adopted by Messrs. Maudslay, which is ill strated by Fig. 148. The elevation shows four spur wheels in gear, two of which are fixed, and the remainder loose on the shafts. The link is a fixed curve formed with each bracket supporting the valve cranked shaft cranks being preferred in the place of eccentrics to impart the motion to the valves. The main, or first motion wheel is fixed on the main crank shaft; the next wheel in gear and that directly above are loose on their shafts, and the fourth wheel is keyed on the valve motion shaft. Now on motion being imparted by the first wheel to the second, the third and fourth wheels are respectively affected, and thus the constant motion for the valve is occasioned. To stop the engines the valves must be placed at half stroke, and this cause is effected by the

Fig. 148.



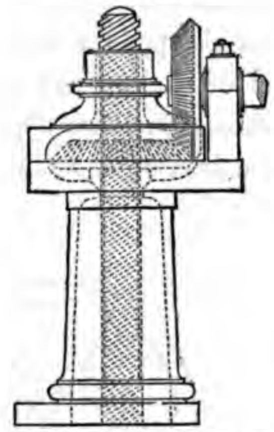
MESSRS. MAUDSLAY'S SPUR GEAR MOTION FOR WORKING SLIDE VALVES.

block in the link being set at half position. The attainment of this is, by the raising or lowering of the levers below—the screwed rod and mitre gearing being the transmittors of motion from the prime mover. It will be noticed that the two loose wheels are supported by the lifting rod, and the lower end of the rod is fixed to a sliding block fitted in the angular groove formed in the framing. The motion of the extremities of the rod is therefore curved at the upper end and straight at the lower. To start the engines, or rather shift the slide valves, the wheel on the valve shaft must be moved, while the lower wheel remains in gear with that on the main shaft. Now it is the constant contact of the teeth that effects this; for obviously when the main shaft wheel is stationary, any motion given to the wheel in gear with it must cause the same to revolve, and thus the two wheels above are likewise turned around. The principle of the arrangement is therefore that the lower loose wheel is the starting, stopping, or reversing agent, and the wheel on the crank shaft the driver. The disposition of the details in plan is compact, and situated centrally of the engines in most cases, rather than as the ordinary link motion, fore and aft of the respective cranks.

STARTING GEAR.

In page 55 a notice is given of the correct place for the starting wheel; it is now the purpose to describe the arrangement of the detail. The practice of Messrs. Penn, Maudslay, and Ravenhill, is to locate the wheel above the engine, either on the condenser or platform, mitre gearing being used to transmit the motion to the screwed rod. The bracket at the link end of the shaft forms an important portion of the arrangement; and an example by Messrs. Penn is seen, in plan, in page 288, and that by Messrs. Maudslay is depicted by Fig. 149. This is a column with a

Fig. 149.

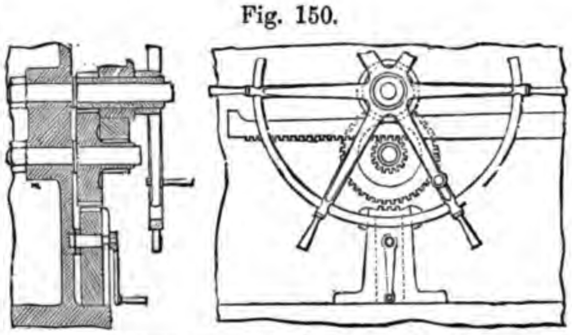


MESSRS. MAUDSLAY'S MITRE GEARING AND BRACKET.

cap suitably formed to support the shafts of the wheels. The application of this will be readily understood by alluding to Plate 19.

Messrs. Humphrys' and Napier's practice is well represented by the Plates 26 and 28, both firms preferring the side of the engine as the best locality for the starting wheel. The

Fig. 150.



MESSRS. HUMPHRYS' STARTING GEAR.

example by Messrs. Humphrys being novel, the illustration, Fig. 150, is introduced to enable its simplicity to be fully appreciated.

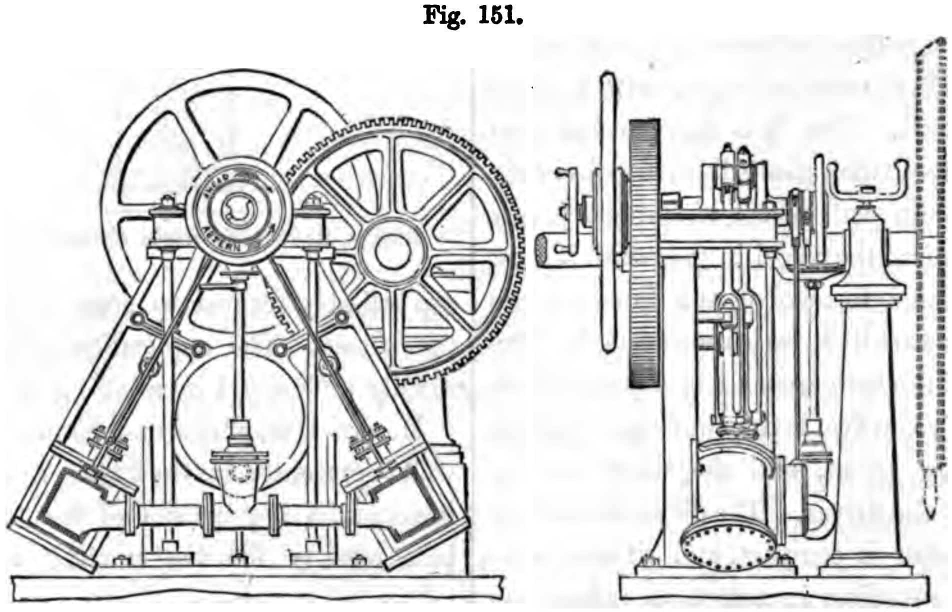
The details consist of spur pinions and a rack —the latter being in connection with the lifting lever shown by Fig. 143 in page 290. Allusion to this arrangement is made in page 56; and, as a conclusion, it may now be added that the gear is locked by the tooth stop, which slides in the bracket specially provided.

Messrs. Watt also put the wheel at the side, or rather end of the engine; their arrangement being shown in Plate 25, consisting of spur gearing, a horizontal screwed rod, and a sliding block. It will be noticed also that, instead of connecting the lifting lever and rod direct, as Messrs. Humphrys, a parallel rod and sliding block is introduced. The utility of this extra detail is, that the arc described by the lever shall not affect the link, resulting from the fact that the extremity of the rod slides vertically during its ascent and descent. The principle of this arrangement is similar to that of the screw and block adopted by Messrs. Penn.

Hand power for raising the link is, after all, a slow process, and for large engines scarcely suitable. It is for this cause that supplementary engines are often introduced, termed steam starting gear. So certain is this addition, that slide valves of 40 square feet area can be shifted instantaneously the power is put into operation.

Messrs. Maudslay have lately designed and constructed a neat arrangement for the purpose under notice, which is represented by Fig. 151.

Fig. 151.



MESSRS. MAUDSLAY'S STEAM STARTING GEAR.

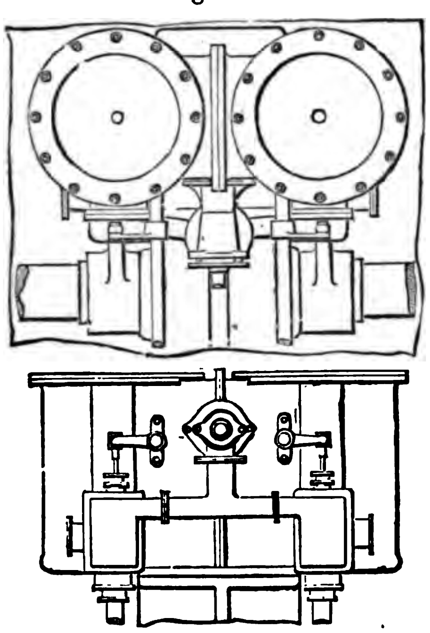
Two direct acting engines are angularly located within a frame of suitable form. The cranked shaft is common to both connecting rods simultaneously, and the power is transmitted to the starting shaft by spur gearing. The arrangement of the supply steam pipe is seen in the end elevation, the regulating or stop cock being between the valve casings. Adjusting discs and stop screws are provided where requisite, and the hand wheel, shown in dotted lines, is only used

when the steam power is disabled or not available. This example has been fitted with the engines, 700 horse power collectively, in the Mail Transport S.S. "Jumna."

Direct acting steam starting gear has been adopted by Messrs. Napier, and the application will be readily understood by referring to Plate 18, where the steam cylinder is directly above the lever in connection with the weigh shaft. This arrangement dispenses with spur gearing, therefore the area of the cylinder is directly proportioned to the power. The available pressure of steam is, however, the principal cause to be considered in the matter.

Other firms have adopted the type of gear now alluded to, and the practice of Messrs. Ravenhill is worthy of notice. Their ideas on

Fig. 152.



MESSRS. RAVENHILL'S STEAM STARTING GEAR.

the subject will be understood by the Fig. 152, being a plan and elevation of a pair of cylinders side by side, the piston rods of both being connected directly to the levers on the weigh shaft. The slide valves are manipulated from the platform, also the steam cock, situated between the levers for working the valve rods.

EXPANSION VALVES AND GEAR.

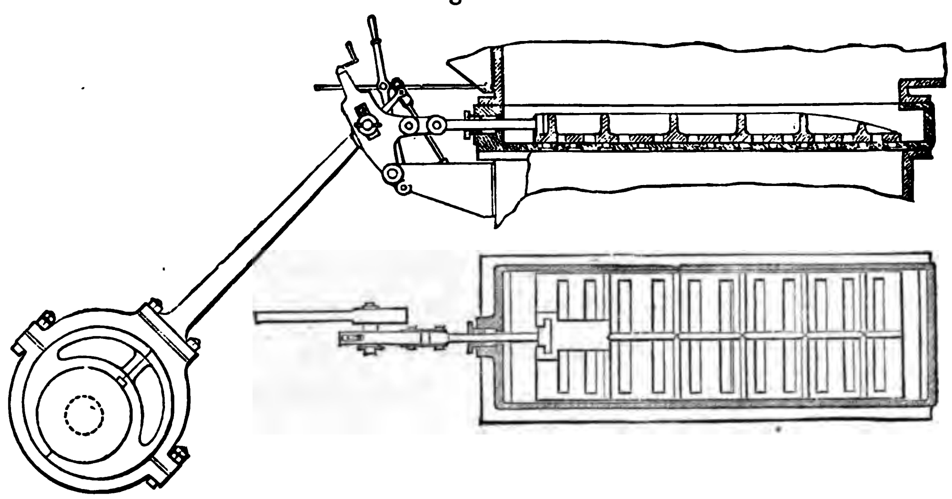
The effect of using steam expansively is treated at some length from page 212 to page 216, inclusive; it is therefore sufficient, as a preface to the present subject, to state that the steam is intercepted between the boilers and the slide valve, by the different modes now to be described.

If simplicity of arrangement is an achievement, Messrs. Penn can claim high honours for their production, shown by Fig. 153—page 298. The expansion valve is the gridiron type, sliding on a seating of corresponding form: both being shown in section and plan, further allusion is scarcely requisite. With reference to the gear, it consists of an eccentric as a prime mover, and the rod is formed with a slot and lift gab at the other extremity. The pin is fixed to a sliding block, which is contained in the grooved portion or link, supported by the bracket at the front of the casing. The cranked handle, above the projection over the groove, is fixed on the screw passing through the block; thus the eccentric pin can be shifted in the groove at pleasure, or to suit the grade of expansion settled on. Suitable means for disengaging the rod and retaining it out of gear is insured by the loop, guide rod, and hand lever, shown above the bracket. The valve rod is connected by flat bars and pins to the link, and guides are obviated by the sufficient strength of the rod and shortness of the stroke.

Messrs. Napier adopt, in some cases, the valve and link nearly as those by Messrs. Penn, at least if not in design, the same in principle of effect and arrangement. This analogy will be better understood by alluding to Plate 28, where the gridiron valve, eccentric, and link, are all shown in sectional and complete views.

The firm next to be alluded to, who adopt the arrangement under notice, is the Messrs. Rennie; but in the place of a gridiron valve, a circular, or rather tubular, perforated valve is preferred, with a seating to correspond. This example is illustrated by Fig. 154, and shows that two tubes—one within the other—comprise the valve and seating, and the difference in the widths of the openings and solid portions between, produce the cut off requisite. The valve slides and derives its motion from an ordinary eccentric, and the change of the grade of expansion is effected by a link and connecting rod attached to the valve rod. Readily to understand the utility of these details, will be to refer to Fig. 155, which is

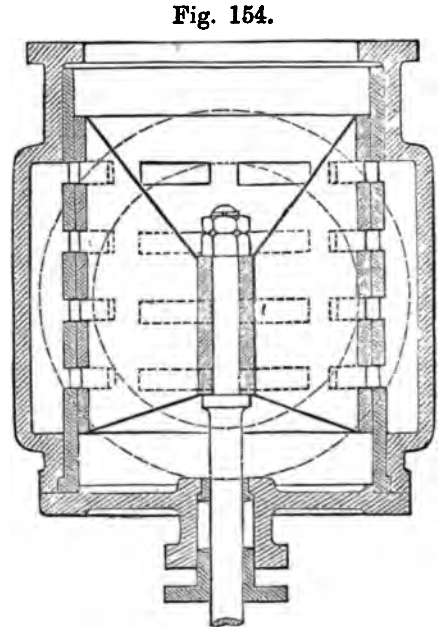
Fig. 153.



MESSRS. PENN'S EXPANSION VALVE AND GEAR.

an elevation of the link and of eccentric rod adjusting handle and quadrant. The con necting rod is also shown at half strok within the slot in the link—the dotted line on the quadrant depicting the utmost move

Fig. 154.

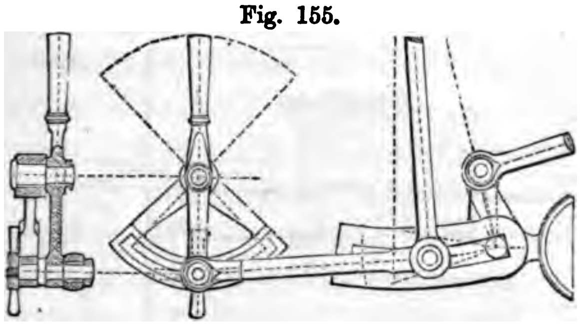


MESSRS. RENNIE'S EXPANSION VALVE AND CASING.

ment, effective or neutral, as required. Th lever, projecting above the link, is keye

on a portion formed with the link, and the counterbalance obviates the weight of the

Fig. 155.

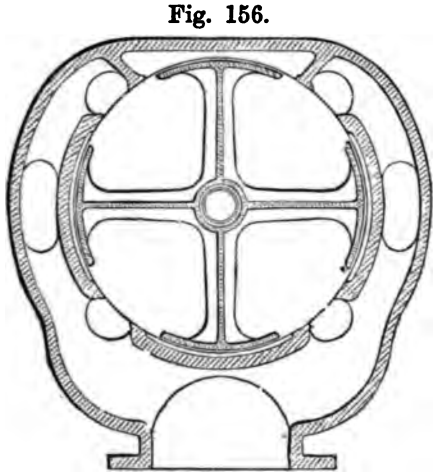


MESSRS. RENNIE'S EXPANSION LINK.

valve and details affecting the motion. The quadrant and hand lever are shown in section, to depict the mode of setting the connecting rod, when the engine is in motion, if requisite. Obviously, the main advantage with the valve in question is, that the pressure of the steam cannot affect the surfaces in working contact, and thus an equilibrium action results.

The tubular valve does not always receive

Fig. 156.



MESSRS. MAUDSLAY'S EXPANSION VALVE AND CASING.

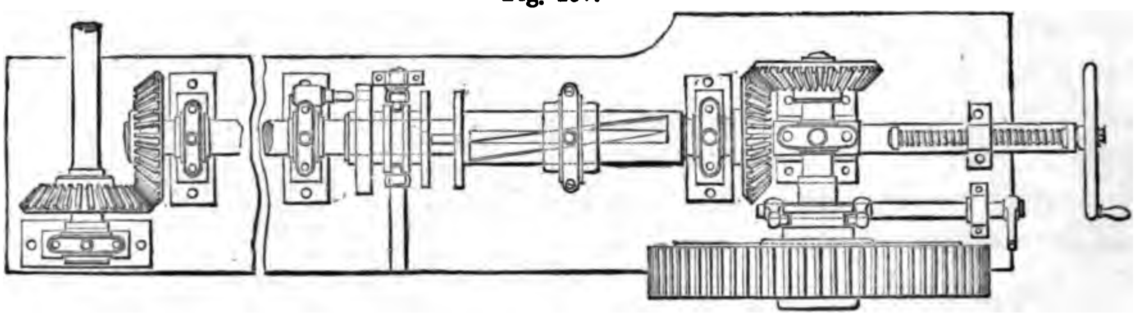
a sliding motion, as the example by Messrs. Maudslay illustrates, as represented by Fig. 156. It is preferred by that firm to cause the valve to revolve within a barrel having openings or ports of less width than those in the valve, by which difference a variable cut off is effected. The motion is derived from the engine shaft, and transmitted to the valve by spur, mitre, and bevel gearing. One of the spur wheels, and the entire set of mitre gearing, is illustrated in plan by Fig. 157—page 300. The position of the gearing in elevation will be better understood by referring to Plate 19. Alluding again to Fig. 157, it will be seen that the counter shaft has fitted on it, between the bearings, a clutch coupling, with discs on each side of the clutch. One of the discs, when in gear with the fixed pin, holds the valve stationary ; and the other, when coupled with that opposite, shifts the valve. As the latter attainment causes the grade or cut off, it will be requisite to further explain the same. On the counter shaft, between the bearing and the coupling, is a grooved barrel, encompassed by a loose ring or boss fitted with a pin, which passes through the slot in the barrel and fits in a straight groove in the shaft. Behind the mitre gearing is seen a hand wheel and screw, which latter is connected to the boss on the barrel. Obviously, then, on motion being imparted to the hand wheel, the boss will advance and recede, and the barrel will be turned in corresponding directions. It is almost needless to add that the pitch of the curve of the groove in the barrel is the main consideration in proportion to the laps of the valve.

Messrs. Ravenhill also adopt mitre gearing for expansion gear; but the valve, instead of being tubular, is square, with curved flanges—the width being proportioned to the stroke of the piston, speed of the valve, and grades of expansion. The casing encloses the valve, and

the rod passes through both details, as illustrated by Fig. 158.

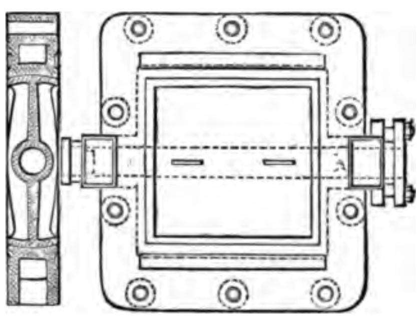
The arrangement of the gearing is simply accomplished as follows. On the extremity of the crank shaft is secured a mitre wheel; a counter shaft—with a wheel at each end—imparts motion to a bevel wheel on the valve spindle. A plan and elevation of the gearing is depicted by Fig. 159. The alteration of the grade of expansion is effected by a grooved barrel, situated directly behind the mitre wheel on the counter shaft. A hand wheel is used to turn the shaft, and a lock nut retains the required position for the valve. This arrangement will be more readily understood by alluding to the sectional view of the same—shown in the elevation. The mitre gearing is disconnected by the clutch and coupling on the extremity of the cranked shaft, and thus the setting of the valve is effected without stopping the engines. The small crank in front of the coupling is for the purpose of working the hand pump when required.

Fig. 157.



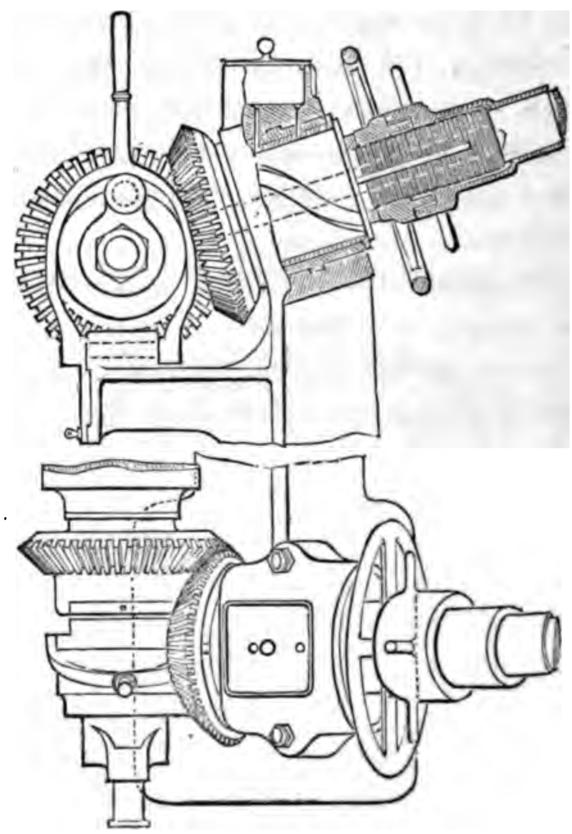
MESSRS. MAUDSLAY'S REVOLVING EXPANSION GEAR.

Fig. 158.



MESSRS. RAVENHILL'S EXPANSION VALVE AND CASING.

Fig. 159.

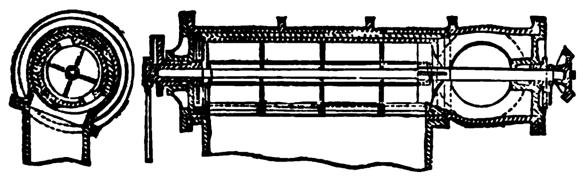


MESSRS. RAVENHILL'S EXPANSION GEAR.

It will be remembered that in all the pre-

vious examples the alteration of the grade of expansion is effected by shifting the valve. Now Messrs. Watt accomplish a similar result by shifting the seating on its axis, the valve

Fig. 160.

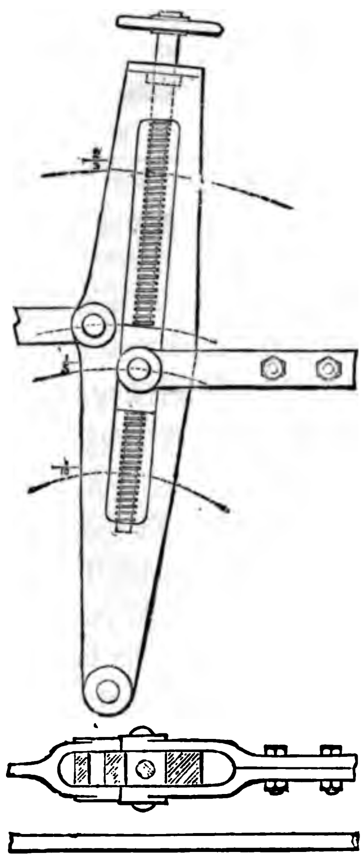


MESSRS. WATT'S EXPANSION VALVE.

being the tubular kind, as shown by Fig. 160. The valve has two ports nearly opposite each other, but the openings in the seating are divided by narrow bars. The seating is supported within the casing, and the valve by the spindle in the centre. The bush that passes through the stuffing box is connected to the seating by an internal coupling, at the extremity of the same; and by the handle on the bush, the seating can be shifted around in the required direction, to alter or set the grade of expansion. The motion for the valve being rotary, mitre wheels with the counter shafts are requisite—one of the mitre wheels being shown at the extremity of the valve's spindle. The application of these valves and gearing for a pair of engines is shown by Plate 25.

Although other firms have had the precedence of notice for the adoption of the link as an intervention between the valve and the eccentric to shift the valve, the credit of the origin of the use of the link for expansive purposes doubtless belongs to T. B. Winter, Esq., C. E., who has used with much success the arrangement shown by Fig. 161. This is a slotted bar, with a screwed rod passing through a sliding block, to which the

Fig. 161.

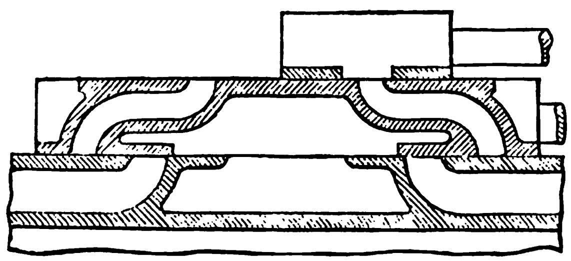


MR. WINTER'S EXPANSION LINK.

eccentric rod is connected. The block is shifted by turning the screwed rod, and its position in the slot determines the grade of expansion. As a sliding motion is available only from an eccentric, Mr. Winter prefers the arrangement of slide valves as represented by Fig. 162—page 302. This is an internal ported valve, with the passages open at the back, on which surface the expansion valve works. This latter valve regulates the admission of the steam, and the former, by the laps, determines the time for expansion and exhaustion. It is almost needless to add

that each valve is worked by a separate eccentric.

Fig. 162.

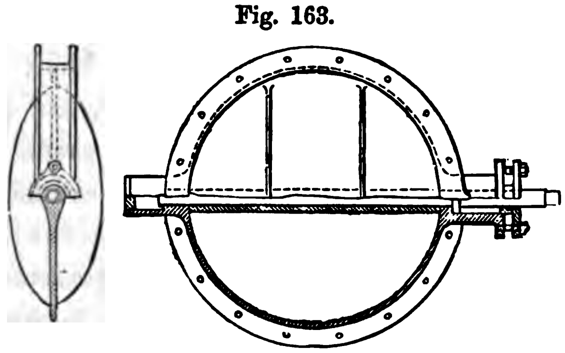


MR. WINTER'S EXPANSION VALVE.

STEAM THROTTLE VALVES.

Independently of the slide and expansion valves, the steam is intercepted beyond them by a third means, termed the throttle valves. This is a supplementary detail used only when independent regulation is requisite, such as starting, stopping, and the boiler's priming. The valve is a disc of metal enclosed in a casing of suitable form, as the Fig. 163 represents—being similar to the practice of Messrs. Penn. The valve swings at right

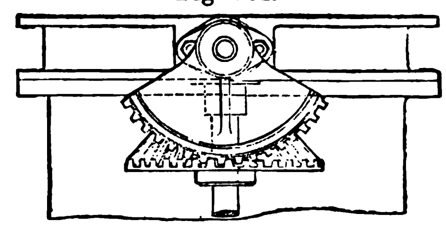
Fig. 163.



ORDINARY STEAM THROTTLE VALVE.

angles to open and close, and the movement is effected by levers. In the place of levers, however, Messrs. Maudslay prefer mitre gearing, as represented by Fig. 164, to move the disc. The same result can be attai by a worm and wheel motion, and also

Fig. 164.



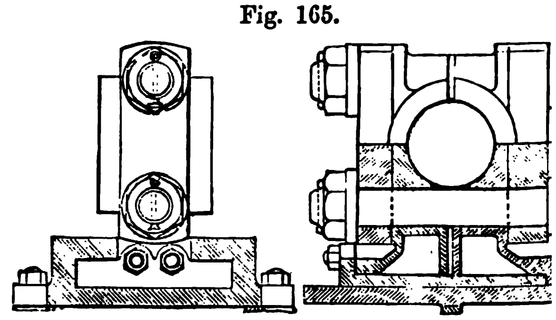
MESSRS. MAUDSLAY'S MITRE-GEAR MOTION FOR THROTT VALVES.

spur gearing. It may be added that action of the handle determines the kind gear motion to be adopted for the val Excepting the valves under notice, Mes Maudslay have often used double beat equ brium valves, which are raised and lowe by levers, or any other mechanical me equally certain.

PISTON ROD GUIDE BLOCKS.

In pages 50 and 51, the advantages of t slipper guide block are made obvious relation to the direction of the crank pi and in page 53, a description is given Mr. Humphrys' single piston ro guide blo

Fig. 165.



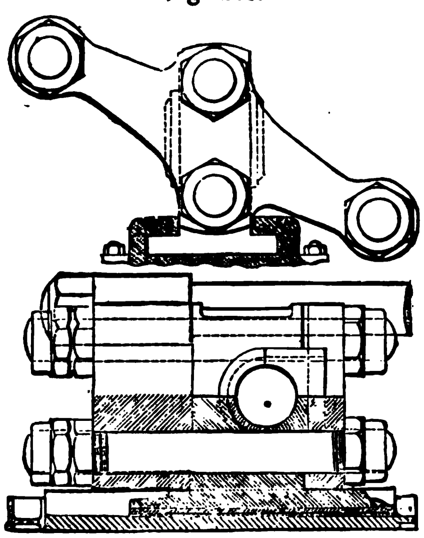
MESSRS. HUMPHRYS' SINGLE PISTON ROD GUIDE BLOC

now illustrated by Fig. 165. The block, already stated, is in halves, each port being almost a duplicate of the other. T

piston rod is attached by the T head, and the securing bolts connect the rod, block, and cap. The slipper portion is separate: the line of contact being angular, adjustment is effected by the set nuts at the front end. The application of this block will be more readily seen by referring to Plate 26.

As far back as the year 1862, we adopted the slipper type for return action engines, attaching the cross-head at the *back* of the block, which was illustrated in our "Practical Illustrations" in the year 1864, and duly referred to in our "Practical Rules" of the same date. Messrs. Napier have lately introduced the same type of block, now illustrated

Fig. 166.

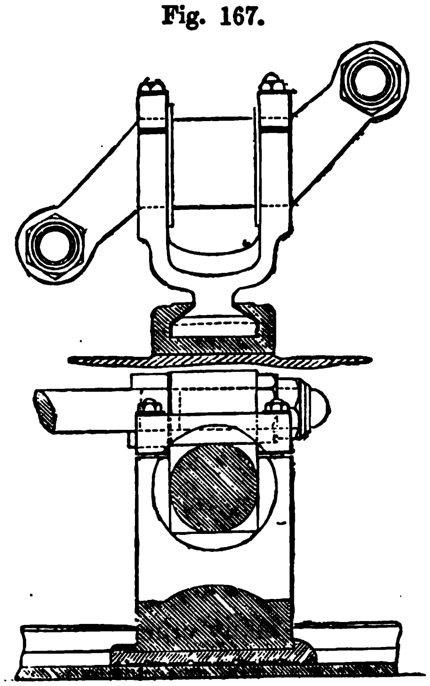


MESSRS. NAPIER'S DOUBLE PISTON ROD GUIDE BLOCK.

by Fig. 166, for return action engines. The cross-head is an angular portion, with eyes at each extremity to support the piston rods; above and below the crank shaft, the rods are secured in the cross-head by nuts. The block is almost solid, the separate portion at the front between the securing bolts being for the purpose of adjusting the brasses when requisite. The securing bolts are fitted at each end with nuts and check nuts, thus equalizing the effect of the strains at the extremities. The slipper is a separate plate into which the base of the block is recessed, with a dovetailed connection to prevent looseness. When the wearing surface of the slipper requires repair, the back end of the same is removed, and the disconnection thus readily effected. The connecting rod for this and the former example is forked, and the pin merely a straight rod. The arrangement of engines in Plate 28 clearly pourtrays Messrs. Napier's mode of applying the block now under notice.

To obviate the forked end of the connecting rod is the aim of many engineers; and Messrs. Maudslay accomplish this by adopting

Fig. 167.



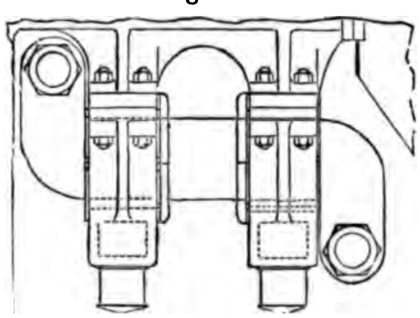
MESSRS. MAUDSLAY'S DOUBLE PISTON ROD GUIDE BLOCK.

the form of block, as shown in Fig. 167, for a return action engine. The cross-head is

angularly formed, the position of the eyes corresponding with that of the piston rods, and the centre part is turned to receive the connecting rod. The block, it is seen, has raised sides, each supporting the cross-head, the connection being effected by the caps, nuts, and studs. The form of the guiding portion is dovetailed or angular at the sides, and the slipper is similarly connected as that in Fig. 166—the adjustment in the present example being by studs acting vertically on the slipper at the ends of the block.

The more universal mode of guiding the cross-head for single-end connecting rods is shown by Fig. 168. This is attained by

Fig. 168.



MESSRS. RAVENHILL'S MODE OF GUIDING THE CROSSHEADS OF RETURN ACTION ENGINES.

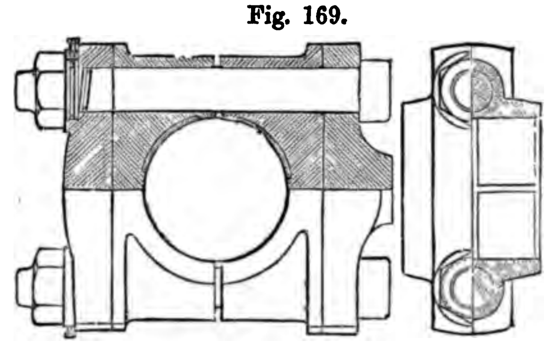
double guides, one on each side of the connecting rod on the centre link of motion, rather than below it, as in the previous examples. The blocks are merely plates of gun-metal with flanges, and suitably connected to the cross-head.

Messrs. Rennie have also adopted this mode of guidance, and many other firms who construct engines of all classes of single and return action. Messrs. Napier have fitted in the Danish armour-plated cupola ship "Rolfkrake" engines with single guide blocks, having the connecting rods—single ends—inserted in them, which will be better understood by alluding to Fig. 8 in page 60, and the description in page 61.

CONNECTING RODS.

In close proximity with the guide blocks are the connecting rods—being, in fact, the main motion details of the engine. The original type of head of the rod was the strap, gib, and cotter. This has, however, given place to the flat brass, and T end, and the semi-solid head. The flat brass head has been brought to its present form by Messrs. Penn, who have developed what the correct propor-

Fig. 169.

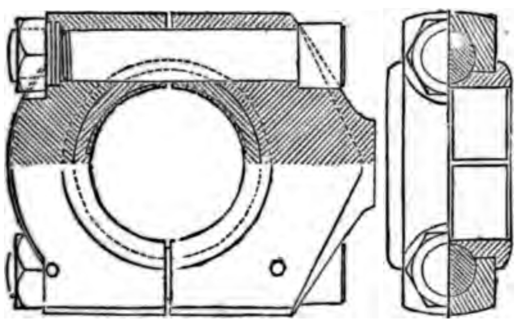


MESSRS. PENN'S CONNECTING ROD HEAD.

tions should be. The illustration, Fig. 169, is partly in section and complete, and shows the design of the firm in question. The securing bolts are as near the bearing as the thinness of the brass will permit, thus reducing the material, while retaining the correct locality for the bolts; that being as near the centre of motion as practicable. The nuts are recessed in the cap, and set screws prevent looseness by bearing in the grooved portion of the nuts. The connection for the lubricator is not seen, but can be understood by referring to the example in Plate 27.

The semi-solid head is so termed from the fact that the cap and the portion on the rod are alike, as illustrated by Fig. 170. This

Fig. 170.



MESSRS. MAUDSLAY'S CONNECTING ROD HEAD.

design is the most symmetrical yet introduced, and the credit of the same is due to Messrs. Maudslay, who have combined strength and elegance in this case. The brasses are circular, and prevented from turning around by the bolts, and laterally by the flanges. The bolts and nuts are retained by the set screws, essentialities with high speed engines. As the example under notice is shown by two views, the remainder can be clearly understood.

Now, with reference to which is the better type of head, the flat or circular brasses, before deciding the question, attention must be devoted to the principle. The connecting rod is the only detail that does receive *three* different motions at the same time continuously in the marine engine. The small or block end is guided by its connection, and thus slides and oscillates simultaneously, while the crank end moves in a circle. As the crank pins are generally large in proportion to their length—for practical reasons, such as forging and turning the shaft, the head of the rod is increased for that reason, rather than any other. It is, of course, certain that the greatest strain on the head is when the rod is at the greatest angle; and, therefore, the side displacement of the bolts are *more* liable at that position. Now, as contact of surface produces stability in all cases, it is obvious that the longer the bolt hole in the head portion formed with the rod, the less the displacement or straining occurs. With the flat brass type the end of the rod is T shaped, and the thickness and the diameter of the bolt about equal; but with the semi-solid head, the length of the hole for the bolt is nearly twice and a half of its diameter. Having thus pourtrayed the advantage of the one type over the other, the conclusion is, of course, in favour of that illustrated by Fig. 170. Most all the marine engineers adopt the two types of heads under notice, as the plates in this work illustrate. The strap and cotter end used by Messrs. Rennie is described in page 45.

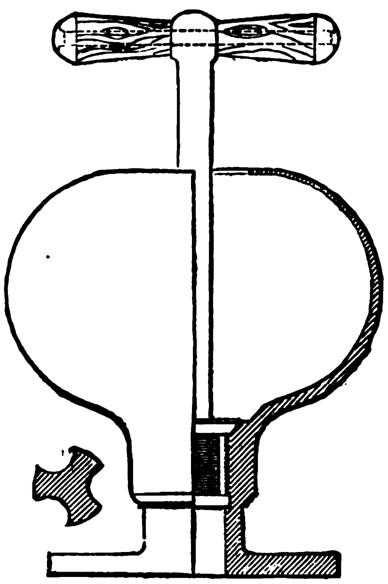
LUBRICATORS.

To reduce the friction of the surface in working contact, lubrication is essential, and particularly so where steam is introduced, or the temperature of the surfaces above the normal degree. The slide valve and the piston are the details requiring lubrication within and connected to the cylinder, and tallow the unguent mostly used in a melted state. The immediate reservoirs are merely cups or cans fixed at the required localities, with stop cocks or valves for the admission of the tallow.

The lubricator for the piston is shown

by Fig. 171, being an ordinary example adopted universally. The cup is almost

Fig. 171.



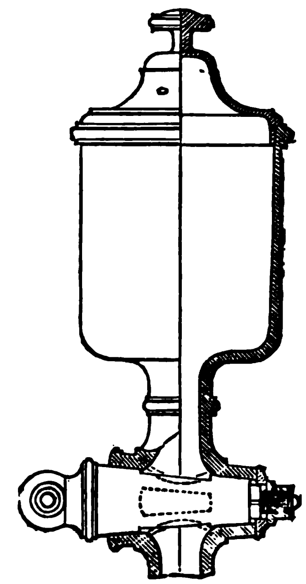
ORDINARY TALLOW LUBRICATOR FOR STEAM PISTON.

globular, and the valve the disc type, with a screwed projection below it as a guide and passage for the lubricant, as shown by the section above the flange. Obviously, on raising the valve from its seat, the tallow can flow through the hollows, and, as the valve is not withdrawn entirely, it can be as readily closed.

In some instances stop cocks are used below the cup, and the form of the latter is not always globular, but often cylindrical, with a fancifully formed cover, as shown by Fig. 172. Lubricators also have been fitted with non-return valves to prevent the waste of the tallow, and others with double reservoirs and cocks for the same purpose.

The continual and economical lubrication of the crank and cross-head pins have often engrossed the attention of engineers, and to the present the suspended can and moti wiper are the best means yet known. A

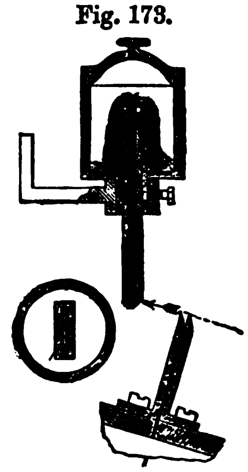
Fig. 172.



PLUG COCK LUBRICATOR

illustration of these details is represented Fig. 173. The can is fitted with a projecti

Fig. 173.



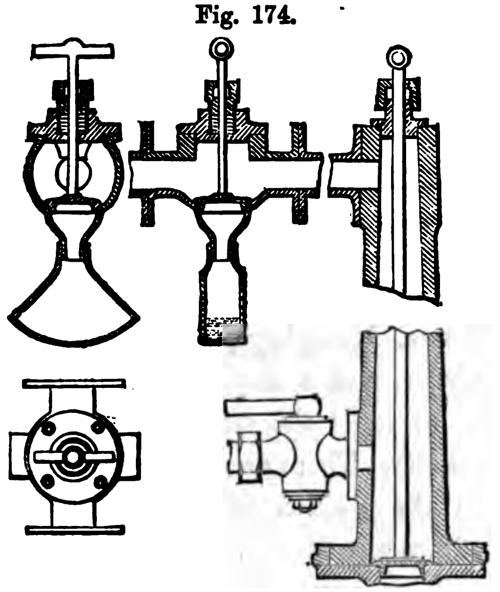
SUSPENDED OIL CAN AND MOTION WIPER FOR CONTINU LUBRICATION.

which contains the extremity of a cert quantity of wick or other suitable mate within the can. The connecting rod l secured on it two pieces of brass, that wi

the wick at each revolution of the crank pin, and thus the use of the oil is regulated accordingly.

When the bearings become heated, which unfortunately they often do, oil and tallow are useless as a lubricant, as the intense temperature of the surfaces and surrounding metal destroy the natural effect of the same when cool. It is obvious, then, that some recourse must be introduced when requisite, and the best yet known is the sea water flowing as a stream or spray over the bearings requiring it. As the adoption of the water is an independent means, separate lubricators are essential, with pipes leading from the "Kingston valve" or the ship side. To regulate the admission of the water, plug cocks are often used; and in some instances valves with handles, to open and close them, extended to the platform, are adopted. An

Fig. 174.



WATER LUBRICATORS AND VALVES.

example of the latter kind is shown by Fig. 174, being an arrangement for the main shaft bearing and crank pin. The tube communicating with the shaft's bearing is supported on the main frame, and the spray cup, shown in two sections, is located directly over the centre of the crank pin. The tubing connects the lubricators. The branch pipe and cock at the side of the vertical tube can be arranged for the main supply, or as an extension to the guide block and channel, and other surfaces liable to heating.

Some firms prefer the oil cups to be cast with, or secured on, the metal above the bearings. The wick is passed through tubes above the level of the oil, and extends to the bearing, the continuous flow of the oil being insured by the contact of the wick and the bearing in motion.

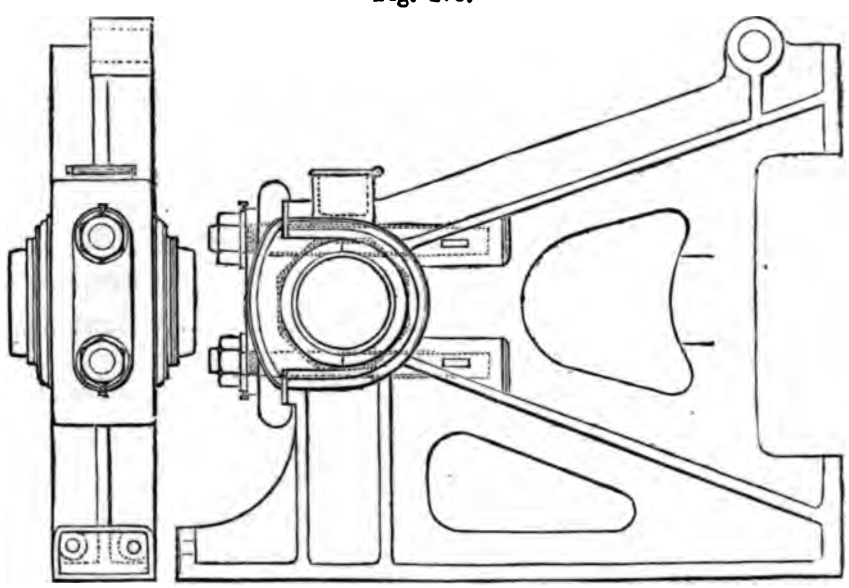
MAIN FRAMES.

The side elevation of an engine owes much of its appearance to the design of the main frame, and doubtless from this cause makers of side lever engines displayed such extravagant notions in days past. In the present period much of the architecture has dwindled into straight lines, and no attempt is now made to imitate any of the orders once held supreme even with marine engines.

The main frame supports the crank shaft, and resists the strain imposed by the motion of the piston rod; and the aim of the designer should be to produce the most simple but correct form to perform these requisitions. As cast iron can be shaped in almost any conceivable form, it is natural that the details in question are mostly made of that material, and an example of the same is shown by

Fig. 175. This has been designed by Messrs. Penn some time ago, and the firm having well considered the same, still adhere to the original form in the main. The frame is connected to the cylinders by the flanges at the top and bottom, and the boss cast on the top rib supports the weigh shaft of the starting gear. The portion of the frame supporting the crank shaft is suitably strengthened by the increased width of the ribs and thickness of the metal enclosing the brasses. The outer bearing of these latter details is octagonal, while in some cases a circular form is adopted. To prevent lateral movement, the brasses have flanges, each being fitted to provisions on the sides of the frame. The cap is supported at the front of the bearing or in line withthe line of strain. The bolts are secured in the frame by keys, and the nuts are inserted in stop rings, with pins and set screws to prevent looseness. It will be noticed also that to relieve the frame from the angular strain much as possible, the cap has clips at ea end, thus clasping the head beyond t securing bolts. The oil cup above t bearing is the ordinary type, and the rema der of the frame can be appreciated from t illustration.

Fig. 175.



MESSRS. PENN'S MAIN FRAME.

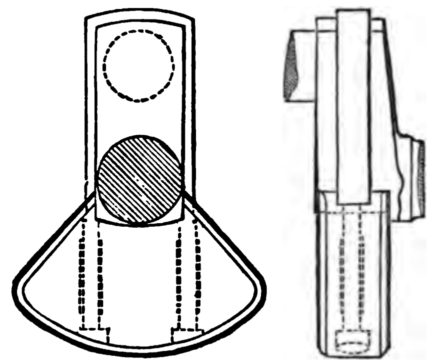
The type alluded to embraces all the requi ments requisite, but even with this acquisiti some makers demur at adopting it. I instance, the form of frame shown in Plate by Messrs. Humphrys, is hollow, or a box girder, and a stay is used in the p of ribs to resist the strain imposed aga the cylinder. There are differences of opir also in the position of the cap, as seen Plate 28, where Messrs. Napier prefer the of the frame as the best situation, and ad side keys to adjust the brasses. Messrs. Rer also used this type some years ago, an exan being depicted by Fig. 4, in page 44. Mes

Maudslay's form of frame is seen in pages 289 and 294, represented by Figs. 142 and 148. The cap is at the front of the bearing, secured at an angle, rather than vertical; the clips also are dispensed with, and the height of the frame is level with the extremity of the cap. The application of the various details now under notice can be fully understood by consulting the various arrangements by the respective firms, illustrated by the woodcuts and plates.

CRANKS AND COUNTERBALANCES.

The manufacture of the cranks and pin is a tedious process, and somewhat scientific. The present practice is to forge the cranks and pin in one solid mass, and the space for the passage for the connecting rod is cut out by drilling and slotting; next, the pin and ends of the cranks are turned, and lastly, the sides of the crank are planed and surfaced. The form of the cranks are, therefore, corresponding with the processes alluded to, by which means nearly all the makers follow a similar design. Messrs. Penn have for some time adopted counterbalances, to compensate for the weight of the cranks and connecting rod affecting the motion of the piston. Messrs. Napier also have lately adopted them, while other firms cast the wheel of the turning gear solid at certain localities. An example of a crank, portion of the pin, and counterbalance, by Messrs. Penn, is depicted by Fig. 176. The balance is secured by a band clasping the end and sides of the crank passing through the balance, and nuts at the extremities of the band complete the connection. The balance is depicted solid; but in some instances it is cast hollow, and filled with lead, to increase the weight to that of cast iron only.

Fig. 176.



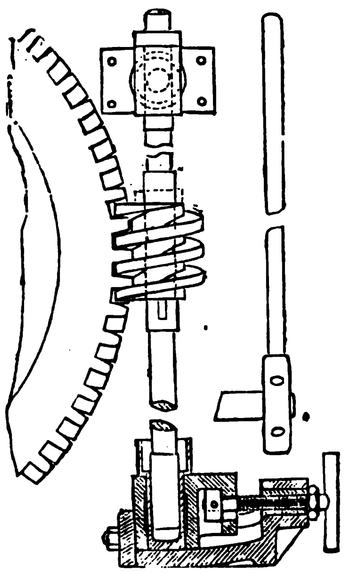
MESSRS. PENN'S CRANK AND COUNTERBALANCE.

TURNING GEAR.

When steam is not available on board, and the engines require repair or overhauling, it is obvious that the pistons must be shifted to examine them; or, should the slide valve require adjustment, its position must be seen at each end of the stroke of the piston, as well as the point of cut off. Apart from this, if the screw propeller requires to be lifted from its coupling, a certain position is requisite. Now it must be remembered that all these duties have to be accomplished by hand power, hence the requisition of turning gear to turn the crank shaft on its bearings. The illustration, Fig. 177, in page 310, is an example of the ordinary arrangement of the present day—being a worm and wheel motion. The wheel is a toothed disc of cast iron keyed on the extremity of the cranked shaft of the engines, as shown in the plates. The worm is fixed on a vertical shaft, but in some instances a horizontal position is adopted. The motion for the

worm is at the extremity of the shaft—being a ratchet brace of suitable strength, and the

Fig. 177.



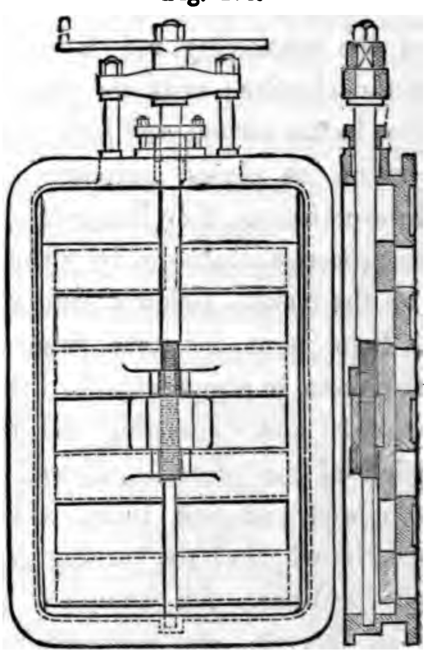
ENGINE TURNING GEAR.

handle about 4 feet 6 inches to 5 feet long. The worm is shown in gear, and when the engines are under steam must be withdrawn, which is effected by the means at the foot of the worm shaft. This is a double bracket, the upper sliding in the lower, and the position required is regulated by the hand screw. The top, or brace bracket, is fixed to the bulkhead or beam of the vessel, and the boss supporting the shaft is secured by a washer and pin, by which connection the requisite movement, when the lower bracket is shifting, is permitted. There are, of course, other arrangements than that illustrated, but the principle of the effect is the same; also the example alluded to is the least liable to looseness or self-disengagement by the vibration of the hull, or the action of turning the cranked shaft.

WATER DISCHARGE VALVES.

The egress of the water from the condenser or bilge is usually at the ship's side. Now, if the discharge were effected through a pipe or clear opening, the ingress of the sea water would be certain when the discharge ceased. It is, therefore, for this cause that discharge valves are situated at the termination of water discharge pipes. The form of valve mostly adopted is the disc kind, with a spindle guide below the valve, and a ring at the top of the rod to raise the valve, which is usually effected by a rope and block. Latterly, the valve introduced is the gridiron type, as depicted by Fig. 178. A novel expedient is

Fig. 178.



WATER DISCHARGE SLIDING VALVE.

introduced to shift the valve. The rod is screwed for a given length, and that portion passes through a block fixed centrally on the back of the valve—better understood by

alluding to the sectional view. On the rod revolving, obviously, the valve will be duly raised or lowered in the casing. The main effect gained with this arrangement is, that the means for shifting the valve is at the correct locality on the valve facing's area, rather than at one extremity, which has been often practised.

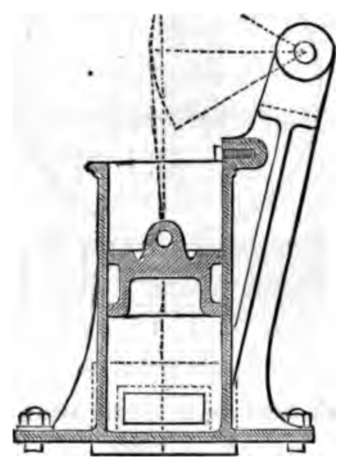
Messrs. Napier have adopted this type—seen in Plate 15. Messrs. Ravenhill and other engineers are also introducing it. The ordinary disc valve casings are shown in Plates 10 and 24, being examples by Messrs. Dudgeon and Messrs. Watt.

FEED AND BILGE PUMPS.

To feed the boilers with water, and to drain the same element from the bilges of the hull, is the purpose of the details now to be noticed. The localities of the pumps and the means of working them adopted by the leading firms can be better understood from the plates of the plans and elevations of the arrangements and sectional views of the engines than any description. It will be noticed that makers of direct acting engines prefer the pumps in question to be cast with the condenser, and the motion derived from the steam piston; excepting Messrs. Watt, who prolong the main pump rods through the back end covers of the chambers to work the pumps under notice, which are separately situated beyond the condenser, shown in Plate 25. Messrs. Napier and Rennie, who use trunks in the air pumps, cast the feed and bilge pumps with the condenser also, and the trunks impart the required motion to the plungers. Makers of return action engines generally secure the pumps beyond the guide channel, and connect the plungers to the cross-heads. Messrs. Ravenhill have obviated this arrangement lately by keying an arm on each of the lower piston rods to impart the motion, and secure the pumps at the side of the channels, as seen in Plate 33.

Steam power is certain when it is adopted, but in the event of its use being prevented, manual or hand power is resorted to, to work the supplementary pumps. An example of this class is shown by Fig. 179. The pump

Fig. 179.



HAND BILGE AND FEED PUMP.

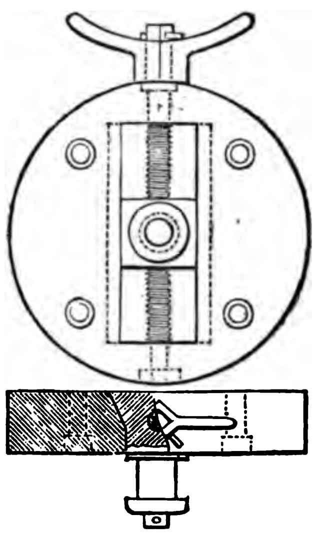
is a simple barrel, with a branch at the lower extremity for the connection of the valve's casing; the hand lever is supported by the projection above the pump, and the plunger or piston is connected by an ordinary rod.

Should steam power be available, a suitable means for returning to it is often adopted to work the hand pump; and an illustration of the same, by Messrs. Ravenhill, is seen by Fig. 159, in page 300.

When the hand pump plunger is required to be disconnected, the connecting rod must be

released, or the motion neutralized. One means of attaining this is shown by Fig. 64, in page 234, in connection with paddle engines, and an almost similar means for horizontal pumps is shown in sectional elevation, in Plate 16. To obviate the hollow rod and key, or set studs, with hand pumps, Messrs. Penn prefer the disc and block arrangement, as shown by Fig. 180. The disc is secured to

Fig. 180.

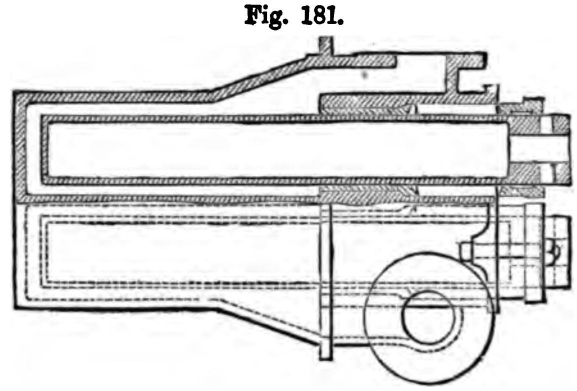


MESSRS. PENN'S MODE OF ADJUSTING STEAM MOTION FOR BILGE PUMP.

the extremity of the main shaft, and thus revolves when the engine is in motion. The block has a pin formed with it, connected to the plunger in the ordinary manner. The regulation of the motion for the plunger, and also obviating it, is effected by a screwed rod, passing through the disc and block, surmounted by a finger and thumb handle. As the block slides in the slot in the disc, it is obvious that its position can be readily shifted, even when the disc is revolving. In the illustration, the centre of the block is on the centre of motion, and thus no motion is imparted to the plunger; but on turning the screwed rod, the block will be raised or lowered, and thus a certain stroke attained for the plunger. The main advantages with this arrangement are, no disconnection is requisite and the stroke of the plunger can be reduced to a minimum.

When the engine feed and bilge pump are in one casting, the arrangement as seen by Fig. 181 is often adopted. The plungers

Fig. 181.



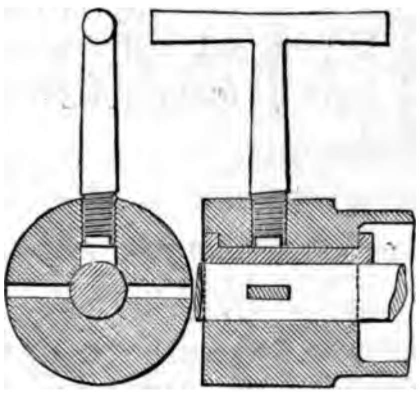
FEED AND BILGE PUMPS.

are connected by a T piece, and thus a single rod imparts the motion either from the piston, piston rods, or cross-head. With direct acting engines, the pumps in question are sometimes inserted in the condenser casting, at the front, below the air pumps; a flange is cast below the branch pipes, as shown in the illustration, and studs and nuts complete the connection. The means of releasing the rod within the plunger is clearly depicted by Fig. 182, and therefore renders further allusion unnecessary.

The most compact, but at the same time correct means of arranging the valves for the pumps under notice, has been duly recognized by the many firms interested in the question. An example worthy of notice is illustrated by

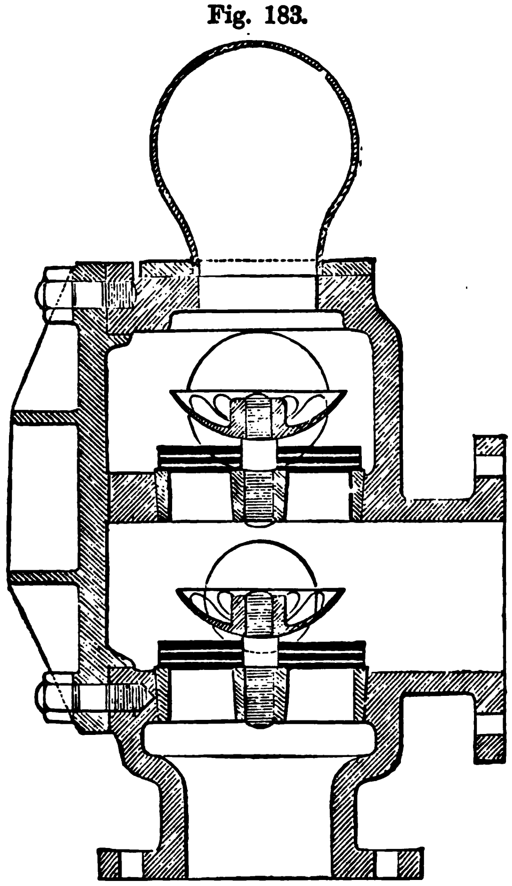
g. 183. One door only is requisite to inspect renew both the valves, and the seatings

Fig. 182.



ET-SCREW AND KEY FOR MOTION RODS OF FEED AND BILGE PUMPS.

Fig. 183.

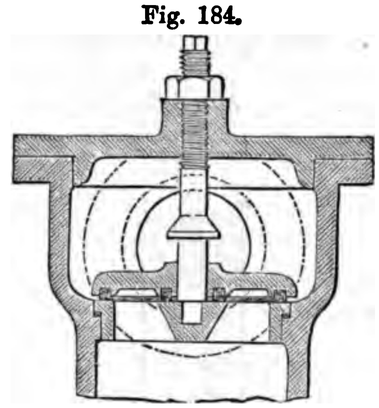


RRANGEMENT OF VALVES FOR FEED AND BILGE PUMPS.

driven tightly in the casing, thus causing a rigid connection. The guards are screwed on the studs, thus dispensing with nuts; the air vessel is above the discharge valve—it is contracted in height, to reduce space—in practice it is extended to twice or even three times its diameter.

The original metal valve and seating cause such a noise, when in action, that even engineers have been unnerved by the incessant "clack;" and to obviate the nuisance, *wood* has been lately introduced as the facing in contact with the seating. An example of this type is

Fig. 184.



WOOD PACKED METAL VALVES FOR FEED AND BILGE PUMPS.

shown by Fig. 184, being a sectional elevation only. The wood is recessed in the valve, and the seating is secured by the bolt in the centre. Messrs. Maudslay have adopted this arrangement with much success, the result being silence of action, combined with a perfect joint with the wood and metal.

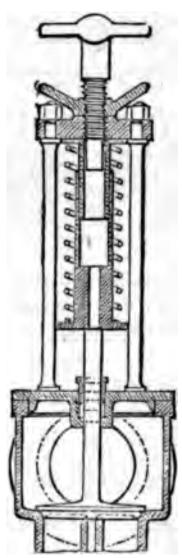
Messrs. Watt also use metal valves, but for bilge pumps only. Each valve is guided below the seating by three ribs, and the requisite lift is attained by spindle guides above and below each valve.

Messrs. Humphrys prefer the valves as depicted by Fig. 108, in page 271, in connection

with the air pumps, for the details under notice at present. Messrs. Rennie have often used ball metal valves for quick motion pumps, and Messrs. Penn the ordinary disc metal valve; while Messrs Ravenhill prefer the india-rubber valves, as arranged by Fig. 183.

All feed pumps must be fitted with relief valves, and the different examples already illustrated for steam cylinders convey a correct idea of the principle for the present purpose. The best arrangement for regulating the spring is by Messrs. Penn, represented by Fig. 185. The spring encloses a telescopic

Fig. 185.



MESSRS. PENN'S FEED PUMP RELIEF VALVE AND SPRING.

spindle, and the set screw and check nut regulate the compression requisite to increase the pressure on the disc metal valve below.

The bilge pumps, excepting the valves, have been explained by the description of the feed pumps, both being similar in design and construction. The valves are usually, as stated, of gun-metal discs, rather than of india-rubber; but in some instances the latter material is still adopted. The suction pipe of the pump in question is fitted with a leaden box at the extremity in the bilge; the box is perforated, and thus any solid matter is prevented from stopping the passage of the fluid.

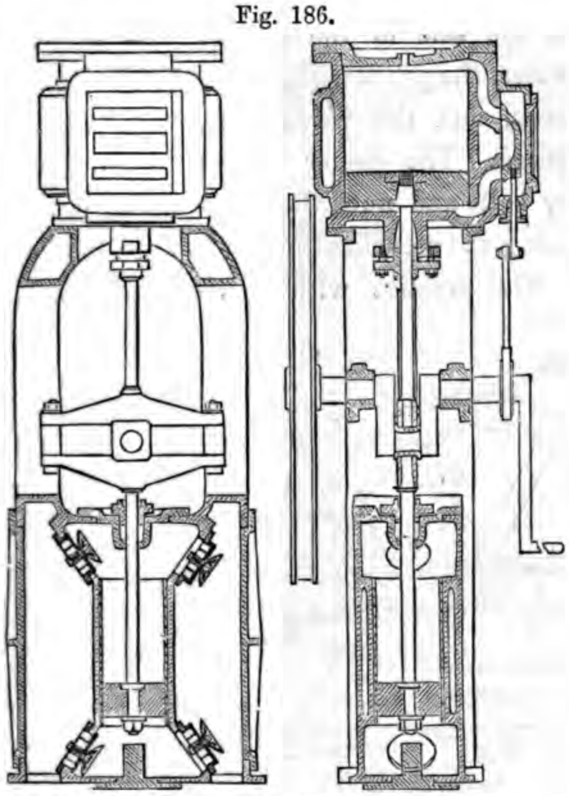
DONKEY ENGINE FEED PUMPS.

The term donkey, applied to these details, doubtless arose from a similar source as horse power; and it probably was considered that any cylinder whose dimensions were below one horse power, the engine must be designated a donkey engine. Now, admitting the veracity of the term for small feed engines, it will not always be applicable in the present day, for the pumps are often 6 to 8 inches in diameter, and the steam cylinders twice the same, the stroke being 10 to 12 inches in length.

The arrangement of the pumps and cylinders are opposite each other, vertically or horizontally secured on the framing. A very compact and neat disposition of the detail has been effected by Messrs. Watt, illustrated by Fig. 186. The cylinder surmounts the framing, and the pump forms a portion of the latter at the opposite end. A duplicate set of valves are secured at the sides of the barrel, thus producing a double action for the piston. The motion for the pump piston is direct from the cylinder's piston, and the return action is ensured by the cranks and fly-wheel. The crank pin is enclosed in a block, and the latter slides in a slotted cross-head, which connects the rods of the pistons. The motion of the cross-head

is vertical, while that of the pin is circular, and thus a continuous action is maintained.

Fig. 186.



MESSRS. WATT'S VERTICAL DONKEY ENGINE FEED PUMP.

The slide valve of the steam cylinder derives motion from an eccentric on the cranked shaft, and the handle beyond is used only when steam power cannot be applied.

Horizontal arrangements are not as universal as the vertical types—doubtless due to the longitudinal space required in the engine or stoking room. Messrs. Dudgeon, however, prefer the horizontal donkey engine feed pump as arranged by them, illustrated by Fig. 187—page 316. The principle of the action of the steam piston, slotted cross-head, crank motion, and fly-wheel, is the same as already described; and, as the cylinder and pump are illustrated by three sectional views, the following description only is requisite. The pump valves are the clack kind, faced with india-rubber, and the guards at the back ensure the return of the valve. The pump is double acting, and the positions of the valves will be better understood from the plan, where the seatings are shown in dotted lines, and the discharge and supply passages in the sectional elevation. Apart from the adoption of the clack valves and the arrangement of the water passages, there is no novelty in the example under notice demanding further attention.

Messrs. Penn, Maudslay, Rennie, Ravenhill, and Napier, in common with others, construct the details in question, and, excepting minor portions, the arrangement by each maker differs but slightly. For instance, one firm may adopt metal disc valves; another, those of india-rubber; and a third may use a legitimate connecting rod; while a fourth prefers to put the fly-wheel at the side of the pump. A very truthful idea of the practice by the principal authorities can be gathered from the plates illustrating the steam launch engines.

Exceptional, however, to this, many donkey engine pumps are constructed without disc or clack valves. The pump is fitted with a slide valve, and passages are cast to correspond. The water takes the place of the steam, as in an ordinary cylinder, and thus the discharge is through the exhaust opening. The slide valves on the cylinder and pump are connected direct by a looped rod, and a pin on the slotted cross-head imparts the motion. This arrangement has been proved worthy of adoption in a few instances, and

there is no cause why it should not receive more attention for further development.

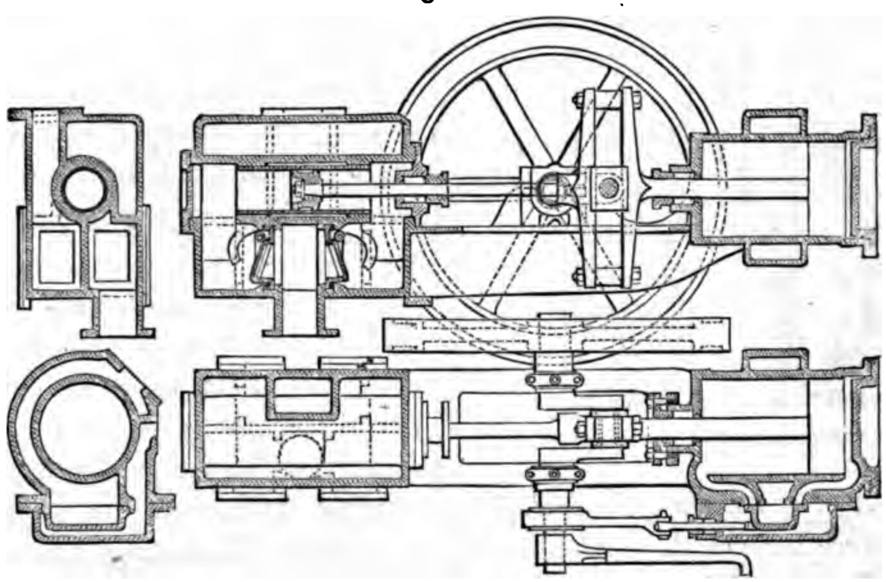
"KINGSTON" VALVES.

Attention is now directed to the concluding detail in the engine room; and, although the last noticed, is the first to be fitted to the hull, long before even the launching is contemplated. The valve in question is prefixed by the name of the inventor for its term, but the utility of the detail is to admit the sea water, for the injection to the condenser, feeding the boilers, cleansing the bilges, cooling the ashes in the stoking room, and the several requirements throughout the ship.

A sectional elevation of the valve and casing is represented by Fig. 188—page 317—also the plan of the guard and guide for the valve. The form of the valve is conical, and the guard is a perforated disc, to prevent the admission of seaweed, dirt, &c. The position of the valve is closed, as illustrated, and the spindle is locked by the clutch above the stuffing box. A plan of the clutch is shown at the side of the casing. To admit the water, the pin is withdrawn, the clutch turned back, and the valve pushed towards the guard. The casing is secured to the hull by bolts and nuts, the flange at the extremity being the securing portion to form the joint.

The practice with wooden hulls was an extension in the length of the casing, and the branch portion was screwed on the remainder, to make the joint secure through the ship's side.

Fig. 187.



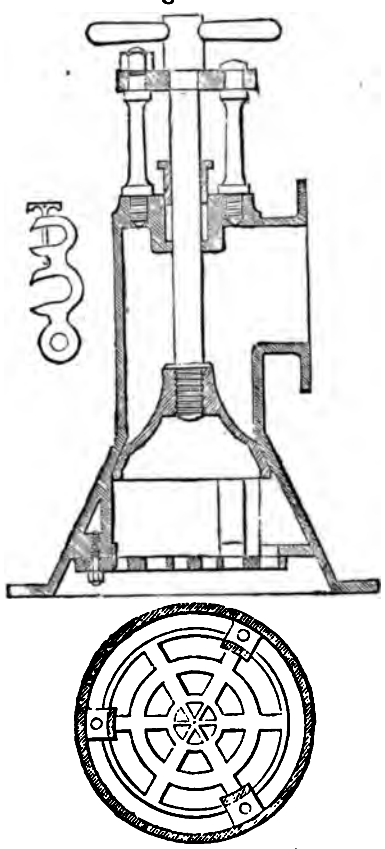
MESSRS. DUDGEON'S HORIZONTAL DONKEY ENGINE FEED PUMP.

THRUST BLOCKS AND SCREW ALLEY FITTINGS.

It is, of course, a well known fact that the screw propeller pushes or drags the hull to which it is fitted, according to the direction it revolves. Now, the shaft of the screw is similarly affected, and the details next to be

are for the purpose of resisting the ;" of the propeller, and supporting the

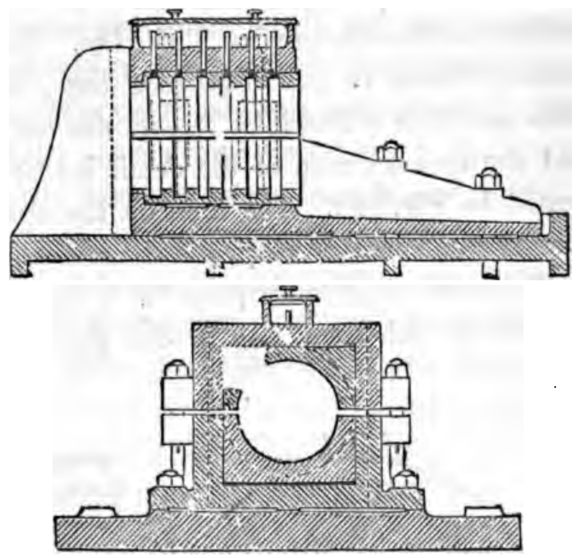
Fig. 188.



"KINGSTON" VALVE.

hence the title thrust block. Obviously he area of the blades and pitch of the of the screw determine the sectional e requisite. The most universal means d to accomplish the desired effect is a of rings forged on the shaft, and brasses o correspond are secured in a plummer of suitable form, of which arrangement 89 is an illustration. The seating of the s is square, as depicted, but in some ces a circular or hexagonal form is pre- , while fitting strips, stop pieces, and s at each end are often introduced. To ensure that the surfaces in working contact are continually lubricated, tubes are secured

Fig. 189.



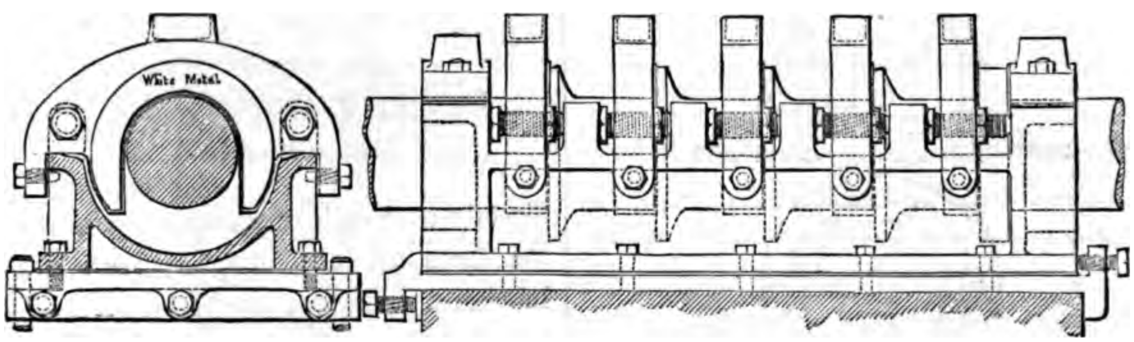
ORDINARY THRUST BLOCK.

directly over each ring on the shaft, and thus the faces are also supplied with sufficient lubricant. The cap of the block is adjusted by bolts and nuts, and the seating secured to a foundation plate, with ribs at the front end to resist the thrust forward; the reverse action being met by a key and a lateral rib at the foot of the plate and block. The ribs below the base of the plate fit into hollows formed in the support connected to the hull, and thus a second resistance is attained, to prevent a shearing strain on the securing bolts. This arrangement is the practice of the principal firms in England and Scotland. Minor deviations of course occur, but, in the main, the design depicted is a correct illustration of an universal example.

Returning to the cause of the effect of thrust blocks, will be to remember that the line of strain is direct; and obviously the

friction is greater on the faces of the rings of the shaft than on any other surface. Now, this known fact generates the idea that adjustment of the resisting surfaces is a valuable addition to the example last alluded to. The accomplishment of this requisition has some time ago been attained by Messrs. Maudslay, and the arrangement of the detail is represented by Fig. 190. The rings on the shaft are faced on one side only, and the resistance of the thrust of the screw propeller is by caps in front of each ring. As in due time the lubrication is ensured by the rings working in a certain quantity of the lubricant contained in the channel below the shaft, and oil cups on the caps. The shaft is supported by bearings at each end of the block, and caps of the usual type prevent the shaft from rising. In the event of the direction of the thrust being reversed, the effect is counteracted by a ring forged on the shaft between the aft-end bearing and the adjusting cap, which also ensures a steady motion for the shaft when revolving.

Fig. 190.



MESSRS. MAUDSLAY'S ADJUSTABLE THRUST BLOCK.

surfaces of the caps and rings are worn, the requisite contact is ensured by set studs passing through each cap at the sides, and lugs cast on the block receive the final effect of the thrust. This will be understood by referring to the side elevation. The studs being screwed through the caps and bearing between the lugs, adjustment is certain by turning the studs, and the nuts behind each cap prevent looseness. To further secure each cap, and prevent its rising, stop studs are fitted, and bear under the flange below the lugs. The block is secured by studs to the foundation plate, and adjusted by set studs at the ends above and below the base line. A constant

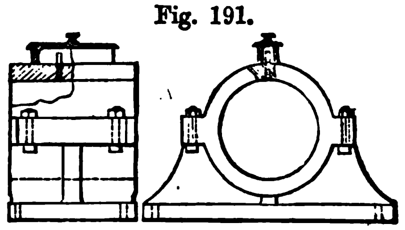
The application of two thrust blocks sometimes occurs, as shown in page 319, by Fig. 192, to resist the strain in either direction, or fore and aft. It is seen also that rings of gun-metal are preferred in the place of solid brasses, as shown by Fig. 189. The lubrication is on the top of each bearing, a hollow being formed in the cap to contain the lubricant.

The screw shaft is supported at intervals in the screw alley by plummer blocks, similar in design as that illustrated by Fig. 191. This is a standard and cap lined with white or gun-metal, while, in other examples, brasses are adopted. The illustration represents the prac-

tice of Messrs. Penn, rather than any other firm.

To render the arrangement of the details in

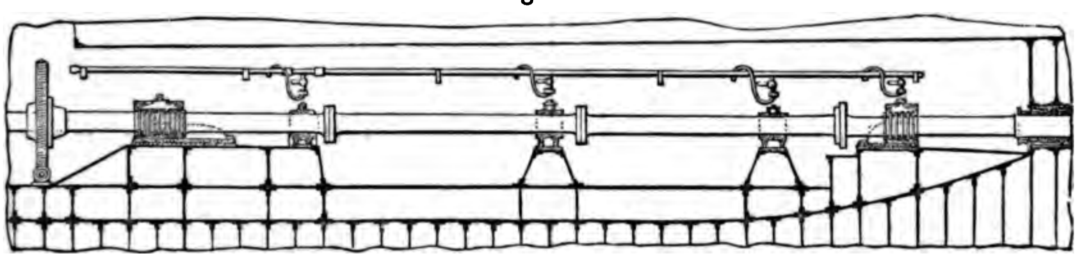
Fig. 191.



PLUMMER BLOCK FOR SCREW SHAFT.

a screw alley and the means of connecting the shafting obvious, the illustration, Fig. 192, is introduced. This is a sectional elevation,

Fig. 192.



SCREW ALLEY AND FITTINGS.

showing the turning gear, thrust and plummer blocks, supports, shafting, and stern tube. The propeller is keyed on the extremity of the shaft beyond the stern post, thus the requisition of two thrust blocks, as depicted. The shafting is coupled by discs forged on the same, and bolts and nuts secure the connection.

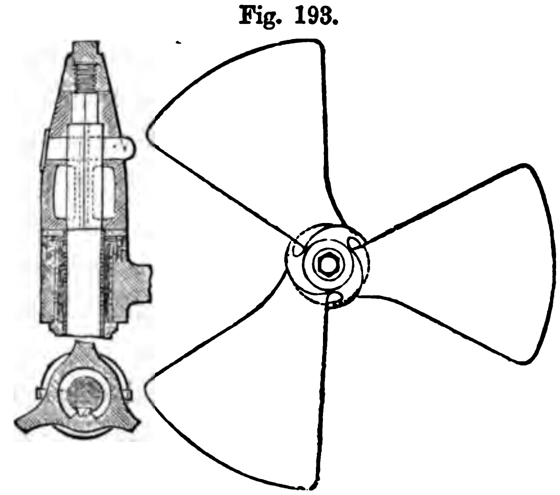
The tubing and cocks are also shown for water lubrication during the occurrence of heated bearings. The practice of Messrs. Penn, Maudslay, Watt, Ravenhill, Rennie, Napier, Humphrys, and the remaining leading firms, is nearly similar in connection with the fittings of screw alleys; and the illustration now alluded to is a truthful representation of actual and general construction.

SCREW PROPELLERS.

In pages 10 to 14, inclusive, a brief notice is given of the types of propellers at present adopted, and those remarks may be considered a preface to this section. As already stated, the "Common, Griffiths, and Mangin" propellers are the practice of the present day, and to these examples further attention is now directed.

The common type has two or more blades—four as a maximum—cast with the boss; the pitch being therefore unalterable, as shown by Fig. 193. This is a three-bladed propeller, often adopted for twin screw propulsion, and is

Fig. 193.



COMMON PROPELLER.

an example constructed by many firms. The

leading corners are curved with a larger radii than those trailing, and thus the vibration is duly lessened. The shaft passes through the boss, and the latter is secured by keys laterally and longitudinally, also by a nut at the extremity. The stern tube is shown, fitted with lignum-vitæ strips, to prevent heating; but in most instances gun-metal bearings are used with examples of ordinary size. Such was the practice of Messrs. Watt for the propeller shafts of the engines shown by Plate 23; and, excepting the lignum-vitæ strips, Fig. 193 is a truthful illustration of one of the propellers adopted. Messrs. Dudgeon also prefer a similar design for the propellers of their engines, and many other firms follow the same practice. The actual dimensions of the example illustrated are: diameter of screw 7 feet 6 inches, pitch of screw 11 feet 6 inches, and length of blade on line of keel 1 foot 10 inches; the diameter of the shaft being 7½ inches, and the length of the boss of the propeller, 2 feet 5 inches.

Messrs. Maudslay, when adopting the common two-bladed propeller, secure each blade to the boss by studs and nuts, rather than cast the boss and blades entire, even with screws 11 feet in diameter. As a conclusion to the remarks on common screws, it may be added that the Fig. 193 is also a truthful illustration of a two-bladed screw by an imaginary opposite location for two of the blades, and the omission of the third.

The correct pitch of the helix of the blade at its extremity is a matter of the highest importance, and no one versed in the subject has illustrated his ideas in practice better than Mr. Griffiths, the inventor of the propeller bearing his name. As experimental evidence is of certain value, the inventor preferred to insert the blades of the screw in the "boss," regulate the angle or pitch by suitable means, and thus be enabled to determine the greatest effect attainable. He also curved the blades forward, to grasp the water on entering it, and thereby gained a proportionate advantage.

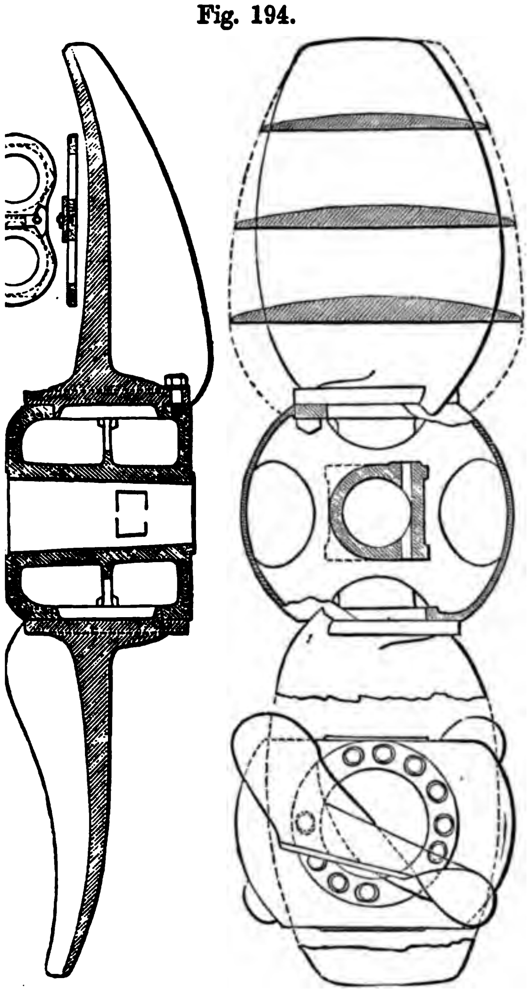
Now, setting aside the form of the blade, the means of regulating the pitch of the screw has been duly considered by Messrs. Maudslay also, by the adoption of gearing, levers, &c., with the advantage of using the contrivance when the propeller was revolving, if desirable.

The Griffiths propeller has been adopted for some time by many firms, and each maker has endeavoured to simplify the connection of the root of the blades with the boss, which is the main attainment. The arrangement of the connection, however, greatly depends on the fact whether "the propeller is to be lifted from, or fixed on, the shaft," which will be understood from the following description and illustrations.

The first example to be alluded to is illustrated by Fig. 194—page 321—being of recent construction by Messrs. Ravenhill for H.M.S. "Lord Clyde," the engines of which are illustrated by Plate 33. The boss is hollow, with a tube at the centre, through which the shaft passes. The connection is made secure by side keys, the position of each being shown in the elevations. The roots of the blades are flanged and secured to the boss by studs and nuts, one of which is shown in the sectional elevation, and the total number for one blade in the plan. To prevent the nuts becoming

, set plates are inserted between them, plate being secured by two studs, as

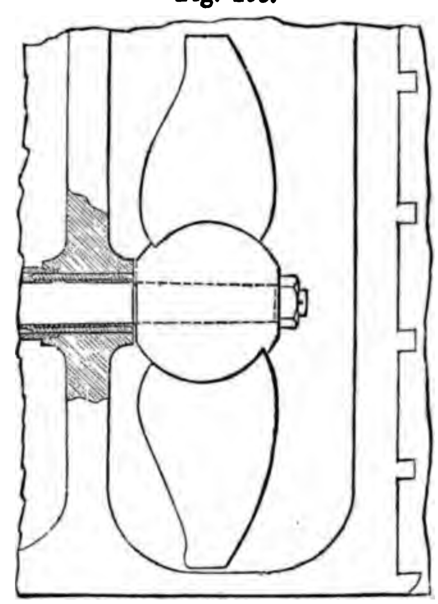
Fig. 194.



SRS. RAVENHILL'S "GRIFFITHS" PROPELLER—FIXED BOSS.

n in detail at the side of the vertical on of the top blade. The form of the full of the blade is shown in dotted lines, transversely of the same are depicted sections at equal distances, thus showthe graduated thickness from the centre he edges of the blade. The form of the ard curve of the blades is shown by the cal sections, also the thickness of the s from the roots to the opposite extremities. The principal dimensions of this propeller being of importance, they are added as a conclusion to the description. Diameter of screw 23 feet 6 inches, minimum pitch 21 feet, mean or central pitch 23 feet 6 inches, maximum pitch 26 feet, length of boss 4 feet 6 inches, diameter of boss 6 feet 6 inches, maximum width of blade 7 feet, minimum width 3 feet 3 inches. To enable the example alluded to to be fully appreciated, an elevation of a propeller stern and rudder posts is illustrated by Fig. 195, being in connection with

Fig. 195.



STERN AND RUDDER POSTS WITH PROPELLER FIXED ON THE SHAFT.

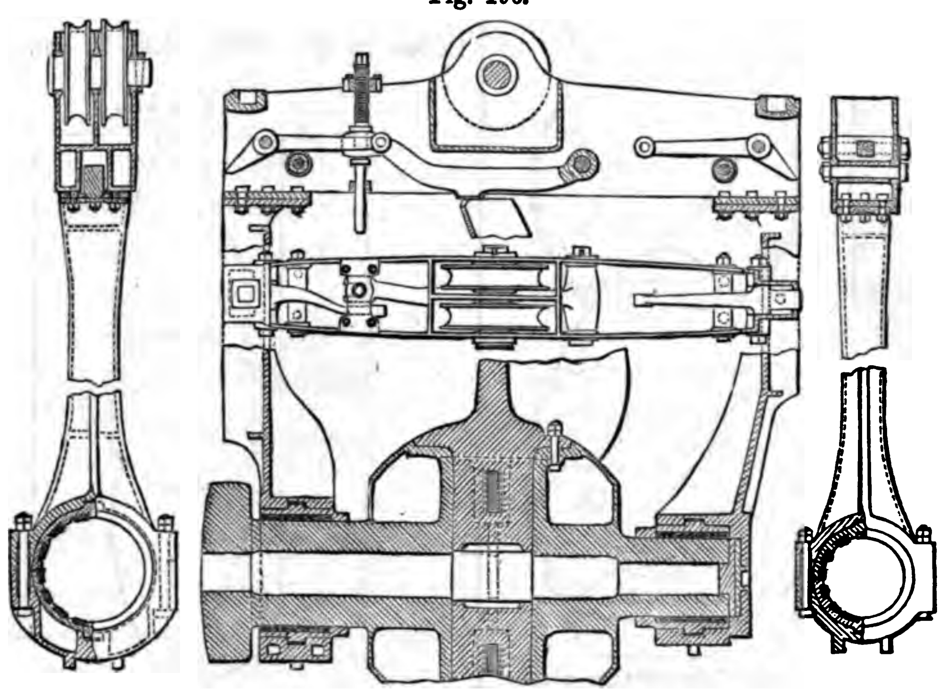
the screw alley depicted by Fig. 192—page 319—thus showing the position of the propeller in relation to the supports for the shaft and rudder.

Should the blades of the screw require adjustment in any of the examples before alluded to, the accomplishment of the same requires that the vessel must either be docked, or raised at the stern when afloat, or the services of

divers resorted to. Now, to obviate these necessities, the propeller is often supported in a frame separate from the stern and rudder posts, termed a banjo frame. This additional detail has been constructed by all the leading firms, and, as the requirements are unalterable for each example, the designs differ but slightly.

The illustration, Fig. 196, is a representation of a portion of a Griffiths propeller and a banjo frame, as generally constructed by Messrs. Penn for propellers about 18 to 20 feet in diameter. The boss and bearing portions are in one casting, and the connection with the shaft is by a cheese coupling; being a disc—seen at the extremity of the bearing—with a projection formed on it, and on the end of the shaft a disc is secured with a groove for the projection to fit into. The bearings are fore and aft of the boss, and fitted with lignum-vitæ strips, the back thrust of the propeller being received by plates at the end of the bearing. The means of securing the caps and connecting the upper and lower portions of the supports will be understood from the end elevations, half sectional and complete. The projections below the metal of the lower portions are stops to fit into holes formed in the stern and rudder post brackets. The sides of the frame are as a box, in section, having the outer sides open. The upper ends are secured to the cross-piece by bolts and nuts, rather than cast with the same, as is the practice of Messrs. Ravenhill and other firms. The catch levers at each end of the cross-piece are for stopping the descent of the frame when lowering, if requisite, or in the event of accident when raising. This is

Fig. 196.



MESSRS. PENN'S "GRIFFITHS" PROPELLER AND BANJO FRAME.

by the sides of the frame being guided
l side brackets, and a ratchet plate formed
ıside; the levers, unless disengaged by
eing always in gear with the ratchet
The central, or stop lever, is used
hen the propeller is required to be
or the steam power dispensed with.
ı is cut in the top edge of each blade,
lever is lowered into the same by the
l rod at the left hand extremity. The
ions and relative positions of the details
under notice will be further under-
y alluding to the plan under the
n of the same.

, as to the means of raising the frame
rew. This is accomplished by ropes
under the pulleys in the cross-piece—
n the elevation,—and connected to
nd sheaves on the upper or lower deck.
se extremity of the rope is connected to
dspike windlass or capstan, and manual
n the power adopted. There have been
es where power from the steam crane
dlass has been used, but the general
is that described.

tain the rigid position requisite for the
when lowered in its seat, and the pro-
n motion, fixing or securing stays are
l, as shown by Fig. 197. These con-
practice, of oak or other timber pieces
to 8 inches square, and 13 to 15 feet
th—the latter according to the depth of
l hole. Each end is tipped with gun-
to prevent splintering. The lower ex-
fits into the recess formed at each end
cross-piece, and the upper or top end
ined by the set screw and bracket, as
nted.

Now the alteration of the pitch of the screw is the main effect, as before stated, and the

Fig. 197.

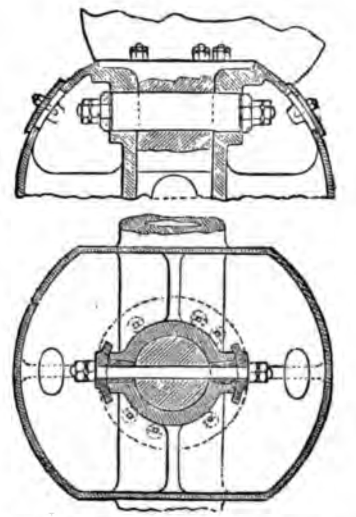
FIXING STAY FOR BANJO FRAME.

mechanical means adopted of no little consideration. It will have been noticed that in the side sectional elevation of the propeller depicted by Fig. 196—page 322—the blade is not only flanged and secured by bolts and nuts, but also a second connection is attained by a key below the flange. A transverse section and plan of this arrangement is shown by Fig. 198—page 324. The key, it is seen, is secured at each end by nuts, and the requisite angle for the blade is retained by wedges, which are prevented from looseness by the washers under the nuts. This mode, although adopted by Messrs. Penn, has been used by Messrs. Ravenhill and other leading engineers.

The principal dimensions of the propeller under notice are: the screw is 18 feet in diameter, maximum pitch 26 feet, central pitch 23 feet, and the minimum pitch 20 feet. The length of the blade on line of keel respectively

to the pitches given, are 4 feet 6 inches, 4 feet $3\frac{5}{8}$ inches, and 4 feet 1 inch; diameter of aft

Fig. 198.

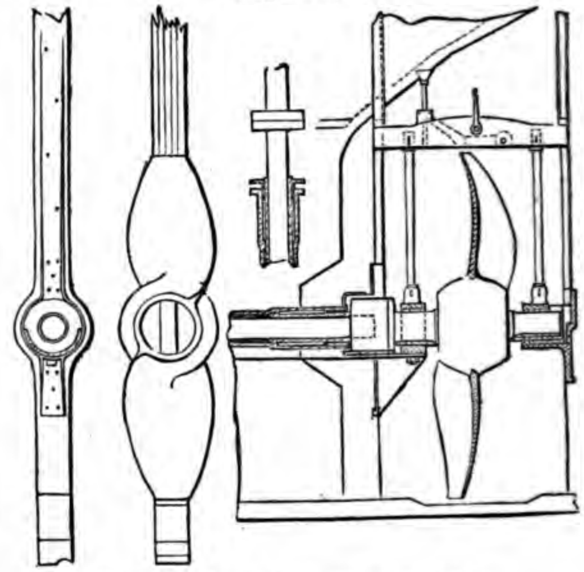


MODE OF SECURING AND ADJUSTING THE BLADES OF GRIFFIFTHS PROPELLER.

bearing 1 foot 3 inches, and diameter of forward bearing, 1 foot 8 inches; length of boss 4 feet, and diameter 5 feet—these proportions being for a pair of engines of 500 nominal horse power collectively.

The lifting frames for propellers, about 8 or 10 feet in diameter, are more simple in form and construction than that depicted by Fig. 196 —page 322. An illustration of a smaller example is shown by Fig. 199. The cross-piece is secured by rods and keys to the cap portions rather than sides forming the connection. The stop lever is manipulated as before mentioned, and the pulleys for lifting and lowering the frame are replaced by a loop, to which the rope is attached direct. The fixing stays are of metal in this case—a common material for the purpose with examples of the size under notice. The stern tubing, stuffing box, and brackets are depicted in sectional and complete views, also a side and end elevations of the propeller. The blades are

Fig. 199.



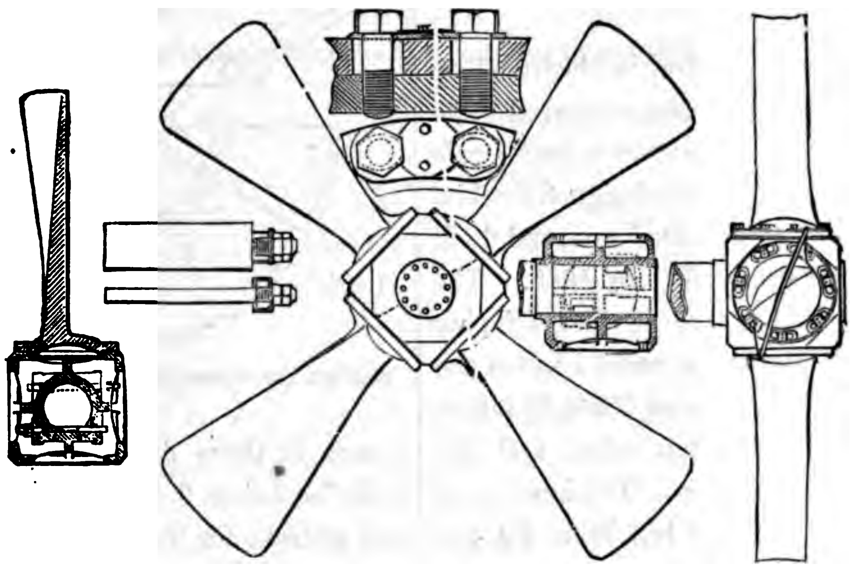
LIFTING FRAME FOR SMALL PROPELLER.

adjusted by key bolts and wedges, centrally located, being similar, as illustrated by Fig. 198. For practical utility, the following dimensions are added: diameter of screw 10 feet, maximum pitch 15 feet, central pitch 14 feet, minimum 13 feet, greatest width of blade 3 feet 4 inches, diameter of boss 1 foot 9 inches, and length 1 foot $8\frac{5}{8}$ inches, width between centres of supports of cross-piece 3 feet 8 inches, diameter of supports $2\frac{1}{4}$ inches, length of set stays 5 feet 7 inches, and diameter of screw shaft $7\frac{1}{2}$ inches. The engines are 150 nominal horse power collectively. This example is the practice of Messrs. Penn, Maudslay, and all the leading firms who have constructed the same.

Having digested the effect and mechanical application of the common and Griffiths propellers, a description of the Mangin type follows, in common with prior notice. This

example being of French origin, it has been adopted more in its native country than in England. Messrs. Ravenhill have constructed a few examples, and Fig. 200 is an illustration of a four-bladed screw, with the Mangin diameter of screw 23 feet, pitch of the leading portion of the blade 23·45 feet, pitch of trailing half 26·55 feet. The mean pitch is therefore 25 feet, but this can be varied from 22 feet 6 inches to 27 feet 6 inches.

Fig. 200.



MESSRS. RAVENHILL'S "MANGIN" BLADE SCREW PROPELLER.

blades, each blade being connected to the boss by studs: these and the stop plates are shown in detail between the upper blades. The shaft passes through the boss,—depicted by the sectional plan,—and is secured by two side keys, details of which are also represented. The thickness of the blade, from the root to the extremity, is shown by the sectional elevation, and the plan of the blade on the opposite side.

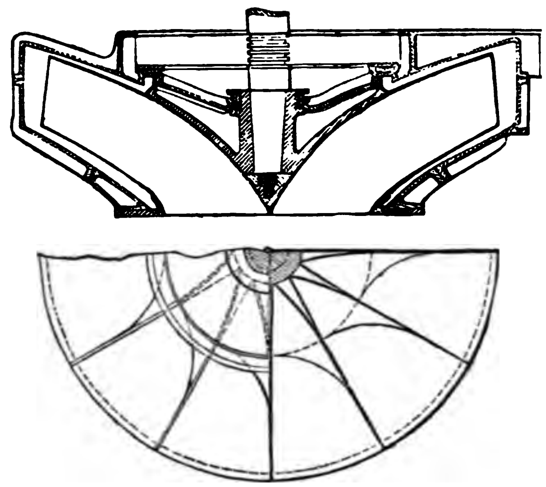
It will be seen that the principal feature of this type is that the leading and following portions of the blade are of unequal pitches, and that the connection of the different helices is at the centre of the blade. The proportions of the example illustrated are:

Notice is now directed to a system of propulsion pertaining more to hydrostatics than mechanical effect, consisting of the action of water when drawn from the sea, and the power attained when forced from the ship's sides above the line of flotation. To accomplish this, a turbine is located centrally of the beam and about midships of the length of the hold. Leading forward from the inlet or casing, a channel is formed with perforations to admit the water—sluice or stop valves being fitted to regulate the admission or prevent the same. On motion, in the certain direction, being imparted to the blades, the water rises and is forced out at each side of the hull. Each discharge pipe has two final branches outside the hull, leading

fore and aft, and the direction of the ship's movement is regulated by the direction of the flow of the propelling agent. This has also been accomplished by pumps in the place of the turbine, but the latter is the better means of producing a continuous stream—one of the main attainments to produce the greatest effect.

An example of a turbine is illustrated by Fig. 201, being of the latest approved form after a series of experiments on a large scale by Mr. Ruthven. This was designed by him and constructed by Messrs. Dudgeon, and fitted in H.M.S.S. "Waterwitch," in 1866. The blades, 12 in number, are 14 feet 4 inches diameter at the bottom edge, and 14 feet at the top; the vertical height being 7 feet 2¼ inches from the inlet to the top edge, and the diameter of the inlet 6 feet. The number of the revolutions attained has been 50 per minute in some instances, and the speed of t ship above 9·5 knots per hour. The moti

Fig. 201.



TURBINE PROPELLER, FITTED IN H.M.S.S. "WATERWITC

power is three direct acting engines, ea cylinder being 3 feet 2½ inches in diamet and a stroke for the piston of 3 feet 6 inche

CHAPTER VII.

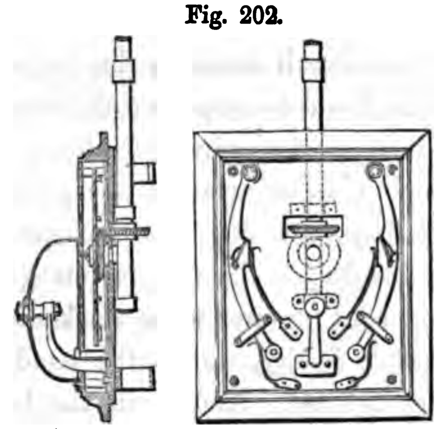
ENGINE-ROOM FITTINGS.

ENGINE-ROOM MECHANICAL TELEGRAPHS, AND ARRANGEMENT OF THE GEAR.

For the proper handling of all steam hips, and especially the quick manœuvring of steam war ships, upon which in naval warfare everything depends, it is manifestly of the highest importance that the officer in command should at all times be enabled to control the motive machinery, so as to regulate the speed in either direction of the vessel, and to be able to at once "go ahead," "stop," or "go astern," according to the requirements or exigencies of the occasion. To attain this end various means have been resorted to, such as a peal of small bells; the use of one bell—a certain number of strokes denoting different signals; and the use of speaking tubes. The two former means are unreliable, and therefore dangerous, on account of the liability to mistake; and the latter has been found absolutely useless, the exact sounds being quite indistinguishable amid the various noises in the engine room and ship inside and the roar of the wind outside.

From what has been thus far stated, it is evident that what is required for the purpose of conveying the signals from the deck to the engine room is not only a means of producing a sound so as to arrest the attention of the engineer, but that having arrested his attention through the medium of one of his senses, another sense should be appealed to to indicate exactly what he is to do. The late Mr. A. P. How early saw the matter in this light, and produced the first successful engine-room mechanical telegraph. This instrument, in its latest form, is now combined with the improvements and modifications of Mr. Gathercole—Mr. How's successor—and represented by Fig. 202, in sectional and complete elevations.

Fig. 202.



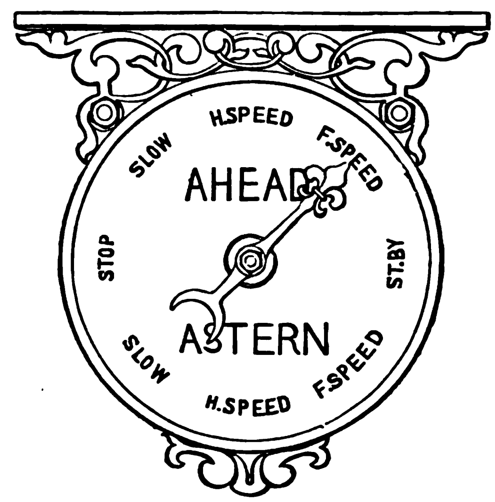
HOW-GATHERCOLE TELEGRAPH.

The arrangement is a fixed front plate having a slot in it for the display of the signals: a revolving back plate carries these signals, and has also at its back studs for the purpose of operating the double striking arrangement, with its springs, as depicted in the complete view. The position of the striking hammer

within the gong, and the whole of the apparatus, will be understood from the sectional elevation.

Engine-room telegraphs, as a means of communication between the officer in command of the ship and the engineer in charge of the engines below, having become now almost general, superseding all other expedients, it is important that the respective arrangements of the signals upon deck, and below in the engine room, as well as the means employed for their transmission, should be reduced to the simplest form. The fact that an engine moving full speed ahead has to pass in succession and in perfect order, although, perhaps, very quickly, through all the different degrees of speed, and even to stop before it can assume a retrograde motion; and again, that an engine moving backwards must pass in complete order through all these stages, before it can be brought to advance—renders it necessary that the order of the signals should correspond with the various conditions of speed required. Nothing can be more simple than to arrange the signals in a corresponding succession with the speeds, so that the hand upon the dial is retrograding when the speed requires to be slackened, and to retrograde finally, should the hand upon the dial so indicate. And again, the hand is advancing when the engines are required to advance, until both arrive at their positive or negative extremes—that is, at "full speed ahead," or at "full speed astern." Such an arrangement would be similar to the arrangement of the figures on the dial of a watch. It is simple and intelligible, at the same time consistent with the degrees of speed of the engine, and easily read off and understood. This arrangement is that adopted by Messrs. Soul and Co. in the telegraphs manufactured by them, the dial plate being illustrated by Fig. 203. When the engines are at full speed

Fig. 203.



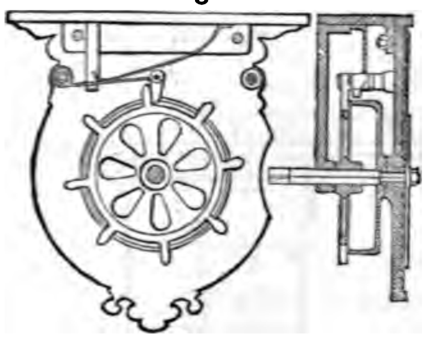
ORDINARY DIAL PLATE.

ahead, the attention of the engineer is called to the telegraph by the alarum-gong. He sees the hand moving in a retrograde direction, and knows at once that speed has to be slackened. He follows the degrees indicated successively by the telegraph with the starting lever or wheel, until the hand comes to a stop, and so *vice versâ*. The hand upon the dial has to pass through the same degree as the starting lever or wheel of the engine, to arrive at any certain point; their direction of action is, therefore, always in conformity. The motion of the handle on deck is transmitted, as usual, by means of a series of shaftings and tooth wheels, where necessary.

One of these telegraphs is represented by Fig. 204—page 239—in complete and sectional views, with the dial-plate—as shown by Fig. 203—removed. It will be seen that the alarm-

mer is directly attached to a spring, and is l upon by a wheel, having on its circum- ıce a series of evolute cogs, the same in

Fig. 204.

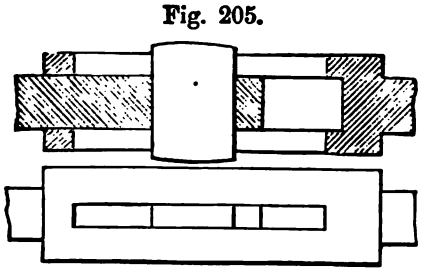


MESSRS. SOUL'S MECHANICAL TELEGRAPH.

ber as the signals upon the dial. To e the sudden release of the hammer, in · that it may strike the bell effectively, a liarly shaped cam is mounted upon it, h, after being raised by the cogs on the l to the requisite extent, allows the alarum ner to fall and to react, without meeting any hindrance which might impair its action should the wheel not be moved : enough. This arrangement dispenses the use of several springs and hammers, ɔut diminishing the number of the required ls.

here great lengths of shafting are required, ɔustomary to connect the lengths by one

Fig. 205.



EXPANSION JOINT FOR TELEGRAPH SHAFTING.

ɪre expansion joints, as shown by Fig. 205, mit of the extension or shortening of the lengths, due to the variation in the temperature, or other cause, and thus insure a proper and easy action of the telegraph under such circumstances.

It is often found that, in mechanical telegraphs in which there is a great length of shafting, or a number of wheels between the operating and indicating dials, the motion of the pointer on the operating dial is not communicated in the same degree to the pointer or other revolving part of the indicating dial; and that there is a certain "lagging behind," or loss, due either to the twisting or torsion of the shafting, or the play, or what is technically termed "backlash," between the teeth of the wheels, and sometimes to both of these causes combined. The effect is, that the pointer on the indicating dial is placed in such a position that it is quite difficult to discover to which of two signals it refers. From the same cause, when a fixed plate with a single slot is used as the front dial, and a revolving plate with the signals engraved thereon is employed as the back dial, it perhaps occurs that only part of the signal is exhibited, or in many cases parts of two signals, thereby creating great uncertainty and confusion, and entirely destroying the useful effect of the telegraph. To obviate these defects, Messrs. Soul have designed two dial plates, as illustrated by Fig. 206—page 330. The signals are engraved on a stationary back plate, and a slotted front plate being attached to revolve on the shaft of the telegraph instrument. The revolving slots permit only one signal, or only one compound signal, to be at any one time exhibited, while at the same time the length of the slots and the space likewise occupied by the letters, comprising

the signal, are so relatively proportioned that the revolving plate may lag behind to a considerable degree, and yet the signal will be fully exhibited.

Fig. 206.

MESSRS. SOUL'S DIAL PLATE, REQUIRING NO POINTER.

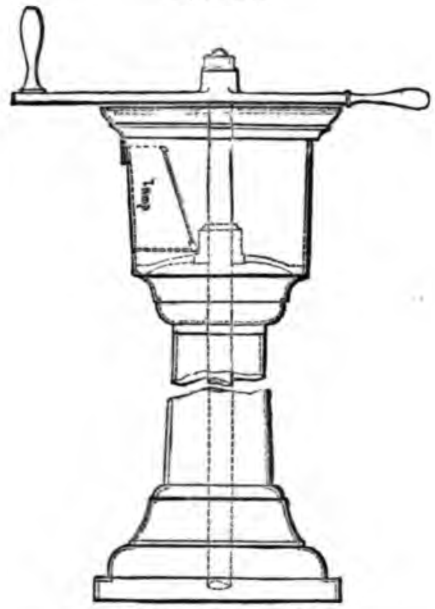
The ordinary means of signalizing from the main deck is by a handle connected to the vertical shaft, a mitre gear wheel at its extremity being in connection with a second wheel secured on a horizontal shaft, and the latter communicating the motion either direct, or by a series of shafts, to the telegraph in the engine room. The deck portion of the complete arrangement is a column, represented by Fig. 207, its height being 3 feet from the foot-plate or deck. A plan of the dial plate or top of the column is shown by Fig. 208. The black portion is brass, and that tinted, of glass, being for day and night utility: a lamp being inserted in the column, as shown, illuminates the glass, and thus the signals are visible.

The complete arrangement of the mechanical telegraphy details, as usually fitted in steamships, is represented by Fig. 209—page 331—being an elevation of the gear, and a section of the decks, girders, and hatchways of the hull.

MARINE ENGINE, SPEED, GOVERNORS.

Marine governors are used for the purp of preventing sudden and violent variations

Fig. 207.



TELEGRAPH DECK COLUMN.

Fig. 208.

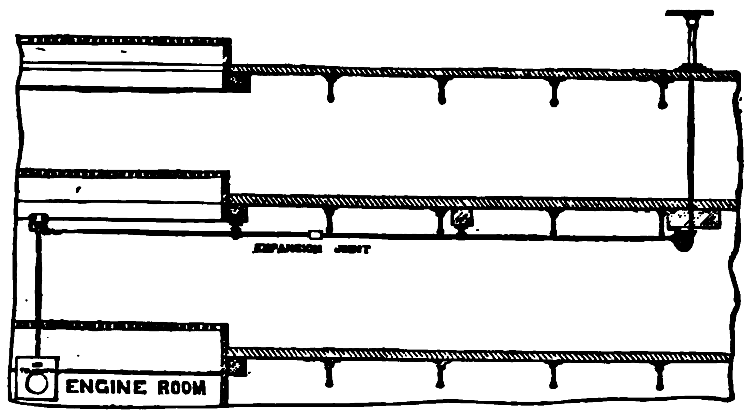
PLAN OF DIAL PLATE OF DECK COLUMN.

piston speed, caused by the emersion of the p

peller. From such variations both engines and hulls of steam-ships sustain strains and other injuries, whilst a very great waste of steam is caused by the engine racing. The changes of speed being so sudden, it will be obvious that in designing a governor for marine engines such an arrangement should be adopted as shall be extremely sensitive, powerful, and prompt in affecting the valve. Modifications of the old two-ball governors were first tried and afterwards balanced four-ball governors. The first failed, as the governing action of the balls was destroyed by the motion of the vessel. With the last, in which the balls are arranged to balance each other in such a way that the action of the instrument is not affected by the same cause, it is apparent that when the sudden acceleration of speed in the engine takes place, the inertness of the balls resisting the sudden motion may prevent the prompt action on the valve.

Attention being drawn to the subject by this conclusion, a governor, consisting of a fly-wheel loose upon a shaft, was next applied for the purpose. There are several varieties of these governors in use, and the following is an explanation of the governing principle on which the action of most such speed regulators properly depends.

Fig. 209.

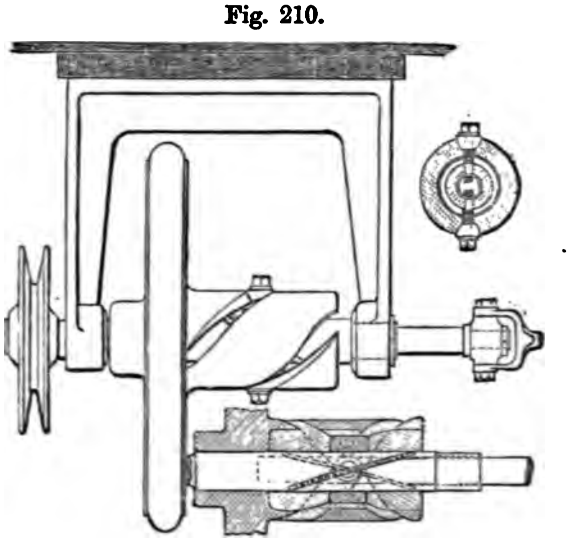


COMPLETE ARRANGEMENT OF A MECHANICAL TELEGRAPH, AS FITTED IN STEAM-SHIPS.

Presume as an illustration that a fly-wheel is fitted loosely on a shaft driven by a connection with the engine, which shaft rotates with a velocity that varies exactly as the speed of the engine varies. The slight friction resulting from the loose fit of the fly-wheel suffices to impart to it in a short time —almost instantaneously—the same velocity as that of the shaft. Some mechanical contrivance or circuit of connection between the fly-wheel and throttle-valve is so adjusted that, while both shaft and fly-wheel rotate at the same velocity—that is, the normal or proper velocity of the engines—the throttle valve is held open. But the instant a sudden increase of engine speed occurs, the relative velocities and position of fly-wheel and shaft are changed, and a differential velocity is created—the revolutions of the shaft becoming a little in excess of those of the fly-wheel—because the slight friction between the shaft

and the fly-wheel is not sufficient to impart, instantly to the latter, the suddenly increased velocity of the former. This advance of the motion of the shaft shortens the mechanical circuit of the connection between the fly-wheel and the throttle valve, and by so operating, closes the valve. Again, for exemplification, when from throttling the steam, or from other causes a diminution of the engine speed occurs, the revolutions of the shaft are a little below those of the fly-wheel, and the circuit of connection lengthens and the valve opens. It is thus evident that the essential means for affecting the valve is, the existence of a differential velocity of the fly-wheel and shaft, and that by this is produced the necessary amount of to-and-fro motion in the mechanical circuit of connection between the fly-wheel and valve. It therefore follows that the more simple the means by which the connection is formed, providing that it be at the same time sufficiently powerful, or capable

Fig. 210.



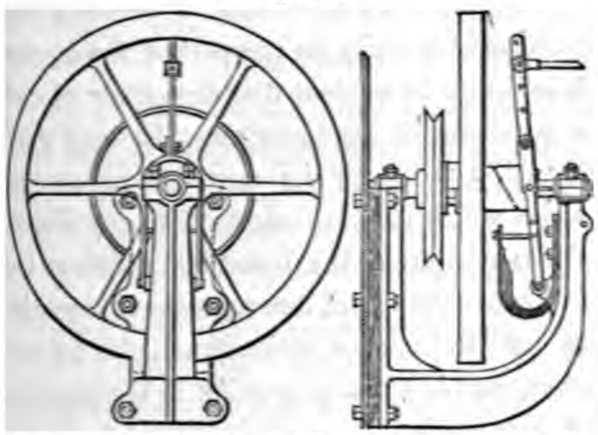
MR. MERITON'S MARINE GOVERNOR.

of easily communicating the result of the differential velocity—the more reliable, sensitive, and efficient will be the regulative action. An example of a wheel governor by Mr. Meriton is represented by Fig. 210. Both the inertia and the momentum of the fly-wheel are taken into consideration, as the governing forces. The contrivance is that, a shaft made wholly or partially hollow, is cut through at portions of its surface so as to form two spiral guides, or double-acting inclines. Upon this hollow shaft is fitted loosely a heavy fly-wheel having at the boss an elongated cylindrical chamber with guides cut spirally through its surface, so as to form also double-acting inclines. In the interior of the hollow shaft is a short spindle attached to a lever for working the valve. This short spindle has a pin passing through it in such a manner that the ends project and form two studs, which fit into holes cut through or in the inner surface of a ring, which ring has also forged upon its outer surface two studs. The two pins or studs within the circle of the ring pass through the spiral guides of the hollow shaft to the short spindle, connected with the lever working the valve; and the two studs without the circle of the ring pass into the spiral guides in the cylindrical chamber forming part of the fly-wheel.

The next example worthy of attention embracing many of the features before alluded to is Messrs. Miller and Knill's governor illustrated by Fig. 211—page 333. The main parts consist of a fly-wheel with an inclined face on the end of the boss; a separate portion corresponding with the inclined end of the boss is loose or slides on the shaft, and is connected to the lever in communication with the throttle valve. The action of the component parts is as fol-

lows:—On an increased motion being communicated to the shaft by the pulley from the

Fig. 211.

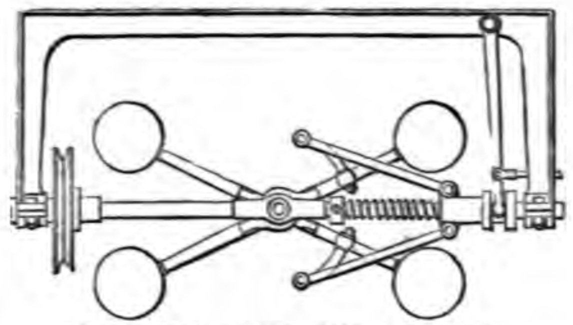


MESSRS. MILLER AND KNILL'S MARINE GOVERNOR.

main cranked or engine shaft, the sliding boss closes the throttle valve. Should the speed of the engine decrease, the spring attached to the lever causes the sliding boss to move in a reverse direction, and thus the throttle valve is opened. It may be added, in passing, that the fly-wheel, although loose on the shaft, is prevented from making more than half a revolution, by centrifugal force, by suitable stops being formed on the back of the boss, and corresponding stops on the boss of the pulley. It is presumed that the extreme simplicity of this arrangement prevents the liability of the details in question becoming disabled.

Mr. Silver's name in connection with marine governors is well known, and the "ball governor" invented by him is illustrated by Fig. 212. It consists of a spindle supporting two arms, which cross each other, and are loaded at their extremities by balls of metal. The arms are connected by links to the sliding collar, and the motion, forth and back, is communicated to the throttle valve by the lever and rod. To insure the return action of the

Fig. 212.

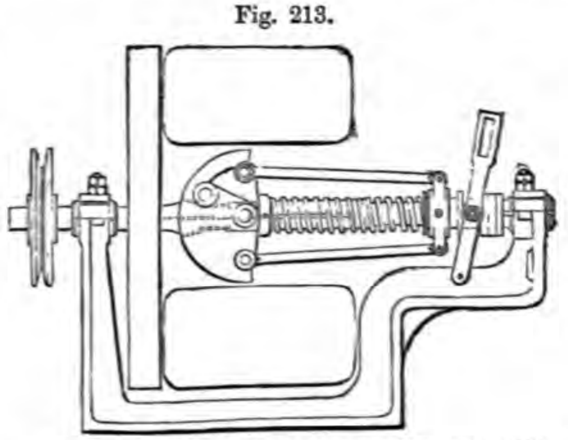


MR. SILVER'S MARINE—BALL—GOVERNOR.

"sliding collar" a spiral spring is wound round the spindle, and a set collar regulates the power requisite by compressing or expanding the coil.

Mr. Silver has also invented another type of governor, termed a "momentum-wheel governor," and illustrated by Fig. 213. This

Fig. 213.



MR. SILVER'S MARINE MOMENTUM WHEEL GOVERNOR.

arrangement is a wheel with four vanes fixed on the boss of a pinion, which works loosely on the spindle and gears into two toothed sectors, these sectors being supported on a cross-head made fast to and supported by the spindle in opposite directions on the pinion;

and, as they are linked by the rods to the sliding collar, a communication with the throttle valve by the lever and rod is certain. The action of this instrument is described by the inventor, as follows :—

"When the spindle of the governor or 'nautical regulator' is turned by the engine to which it is attached, the two-toothed sectors, which are carried on the fixed cross-head, being geared into the pinion on the momentum wheel, have the tendency to turn round on this pinion; but as they are linked to the sliding collar, they necessarily pull inwards this collar, and so compress the spiral spring, and this spring reacting on the collar, and consequently on the toothed sectors, serves to turn round the momentum wheel, while the vanes on the momentum wheel balance the action of this spring by the resistance the atmosphere offers to their progress through it. As the leverage action of the toothed sectors upon the momentum wheel pinion increases, as the spring becomes distended, and *vice versâ*, it will be seen that the reaction of the spring in propelling the momentum wheel will at all times be uniform, and as much only is required as will carry round the momentum wheel with its vanes at its proper speed, and overcome the friction of working the throttle valve, and throttle valve connections. When the momentum wheel is in motion, it will rotate with the engine to which it is attached, at a velocity proportioned to that at which it is fixed by the connecting gear; and while the engine from the usual causes may attempt to vary this velocity, it cannot affect the momentum wheel, but leaves it free to act upon the sliding collar, and consequently upon the throttle valve—at one time closing the throttle valve by its action in resisting any increase of velocity, and at another time opening the throttle valve by its action in resisting any decrease of velocity on the part of the engine. It will now be evident that the power of such a governor or regulator must be very great indeed, having for its agent a momentum wheel which may be increased to any dimensions; and from the powerful resisting tendency of such wheel, it necessarily follows that its sensitiveness of action must also be very great, and in exact proportion to the tendency of the engine to vary its speed; and the engine itself being the direct prime mover of the throttle valve, it also follows that the inert power of the momentum wheel increases its resistance exactly in proportion to the rapidity with which the engine varies its speed."

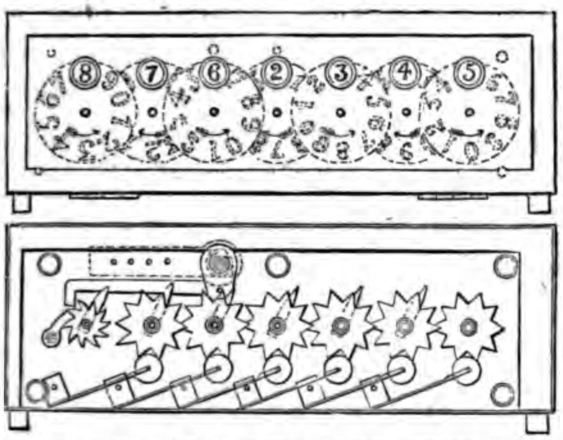
These governors—Figs. 212 and 213, page 233—are manufactured by Mr. Hamilton of Glasgow, to whom the author is indebted for the working drawings and particulars. Thanks are also due to Mr. Meriton and Mr. Miller for their contributions—Figs. 210 and 211, pages 332, 333.

ENGINE COUNTERS.

Amongst the various requirements of the marine engine room, a reliable means of indicating the number of revolutions made by the crank pin is rightly considered to be of no small importance, and a modern engine room would be considered far from complete if unprovided with some description of instrument for this purpose. An example is represented by Fig. 214. It will be seen that a series of disc plates, each bearing the Figs. 0 to 9, are mounted

behind a front plate having slots, arranged in a line in such manner as to exhibit one figure only of each disc plate. Behind each disc plate and on the same spindle, each disc plate has a

Fig. 214.



MESSRS. SOUL'S ENGINE SPEED COUNTER.

ten-toothed wheel. The first, or unit wheel, is worked by a catch lever, on a spindle projecting through the case of the counter, and connected by external levers and rods with an eccentric upon the shaft, whose rate, or number of revolutions, it is desired to indicate. This unit wheel is provided with a paul, and has at its back an arm. The spindle of each wheel projects through the front plate, and has on it a square, so that it can be set by a key. In starting, the discs are all thus turned so as to indicate 0. The engine and shaft having made nine revolutions, and the Figs. 1 to 9 having appeared through the slot when the tenth revolution of shaft is made, the arm on the unit wheel will operate the wheel of the next disc plate, and cause it to present the figure 1, while the 9 on the unit wheel is turned to 0. When ninety-nine revolutions have been performed, an arm on the second wheel will operate the third; and when nine hundred and ninety-nine revolutions have been made, the arm on the third wheel will operate the fourth wheel, and so on; that is to say, for every complete revolution of the unit wheel, the second wheel will be moved one tooth; for every revolution of the second wheel, the third wheel will be moved one tooth; and thus throughout the train of wheels, each wheel but the last being provided with a lever to act upon its higher numerically-powered neighbour. All the wheels but the unit wheel are each held in positions by a spring having at its extremity a small friction roller bearing lightly against the wheels, but exerting sufficient pressure to keep the wheel from moving, excepting when it is actuated by the arm of the adjoining wheel. By these counters can be seen at a glance the exact number of revolutions made; and it is presumed by Messrs. Soul that, in those of their manufacture, the whole of each figure is at once presented, and not merely a portion at a time, as may be the case in some instruments with a nearly similar arrangement; while the peculiar construction of the springs, and the careful proportioning and fitting of the several parts, prevents any inaccuracy from the causes from which many such instruments fail, namely, the jumping of the wheels.

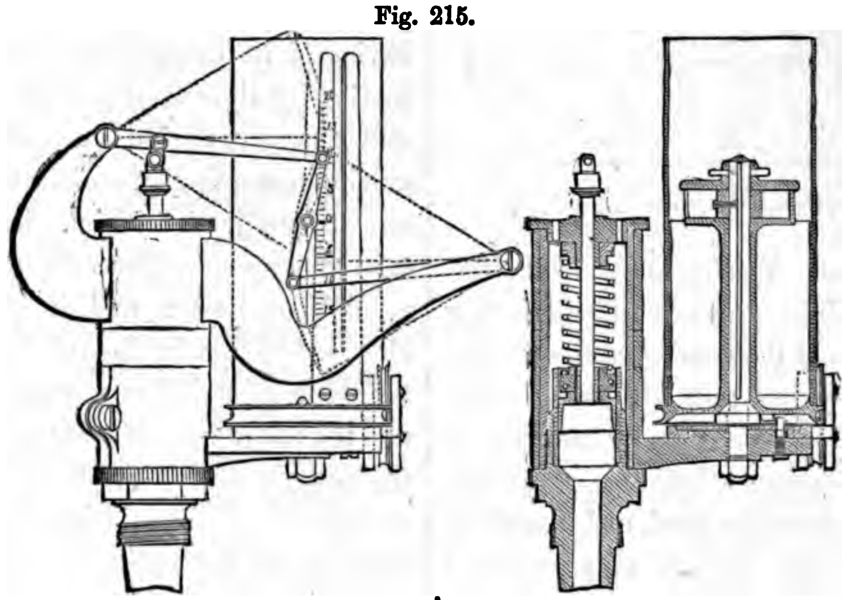
STEAM INDICATORS.

An indicator diagram should be a faithful illustration of the action of the steam on the piston, the control of the steam by the slide or expansion valve, and the points of admission, expansion, exhaustion, and compression. To understand these acquirements, a truthful illustration is requisite, and it is to be deplored that

too often this is not attained, either from a fault in the connecting gear, or the instrument.

The action of an indicator is to describe a figure on a piece of paper, which vibrates at the same speed in principle as the engine piston slides; the steam that is in the engine cylinder is permitted to escape into a small cylinder containing a piston and a spiral spring above it to insure the piston following the steam on the decrease of the pressure. In connection with the indicator piston rod is a pencil, which marks the motion and the speeds of the paper and the engine piston must be alike in "principle." To be concise, these two accomplishments are all that is requisite, as far as motion is concerned, to take a true figure. With reference to the area of the steam pipe and piston, there is not the least doubt that both should be the same, but with the present forms of indicators a difference is, perhaps, advisable, the piston having an excess of $\frac{1}{8}$, or even $\frac{1}{4}$, if preferred. The stroke of the piston should be as short as practicable,

Fig. 215.



MR. RICHARDS' INDICATOR.

of the piston on the paper. Now, evidently from this, the steam should have free operation on the indicator piston, to cause the latter to indicate correctly. Notwithstanding this, however, indicators have been constructed with contracted pipes in connection with the cylinder, and thus an erroneous effect resulted. In other cases the indicator pistons have been made too small, and with too long a stroke, or, in other terms, with high speeds useless. It seemed to be forgotten that the indicator piston must follow the steam on the decrease of the pressure, and the spring at the back of the most sensitive temper, allowing for the compression and expansion. The barrel on which the paper is secured should be fitted with a very strong watch main spring, of equal power throughout the vibration of the paper, and the string or gut connecting the engine motion rod should be well stretched. Last, but not least, the indicator should be connected to the ends of the cylinder—never on the side—as close to the metal as possible.

The best indicator at present known in Eng-

l has been introduced by Mr. Porter, an erican gentleman, being the invention of Richards, also from the same country. It constructed by Messrs. Elliott Brothers, idon, to whom the author is indebted for working drawings depicted by Fig. 215. s is a sectional and complete elevation of an icator, embracing the latest improvements the connections, no screwdriver being re- site to disconnect any portion. The section icts the steam piston, spring and casing, also motion barrel and the spring to cause the ction. The complete view shows the levers nected to the end of the piston rod, and portion containing the marker, which is allic, paper being used of a suitable nature.

It will be noticed that the levers form a parallel motion, thus causing a straight line for the path of the marker. The stroke of the piston is one-fourth of that of the marker, an advantage of the highest importance with high speeds, for which the instrument is well adapted.

The vacuum and steam gauges in the engine room are located respectively on the exhaust and supply steam pipes in a prominent position in the engine room. The gauge mostly adopted is by Mr. Bourdon, being a coil of tubing filled with mercury; the pressure acting on the coil causes the hand on the dial to indicate the power exerted. Steam gauges are also fitted directly to the boilers.

CHAPTER VIII.

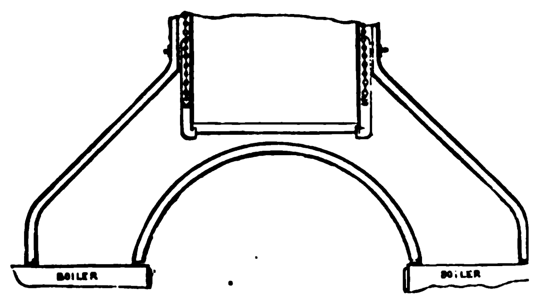
DETAILS AND FITTINGS FOR MARINE BOILERS.

It is well known to engineers who have had practice at sea that the details of the marine boiler require as much attention as the engines; while it often occurs, on the other hand, that people destitute of this information think directly opposite, and too often say, and act it also, that "anything will do for a boiler." Now, with legitimate firms this mistake rarely happens, hence in the main, the details of marine boilers have received from those firms the attention they deserve. It is now the present purpose to describe the many productions, selecting the best examples as a study for the student, and a guide for the young engineer in practice.

UP-TAKES.

These details form a communication from the smoke boxes to the chimneys, and their form depends chiefly on the position of the boilers in plan. To fully understand this, the illustration, Fig. 316, is introduced—being an example lately fitted by Messrs. Watt in H. M. S. "Research." The boilers are arranged so that the stoking space is between them, which is a common practice where transverse space is available. The up-take is curved, and of an equal form in connection with the boilers, by which means an equal escape for the heated products results. It sometimes occurs, h ever, that this form is not applicable, and

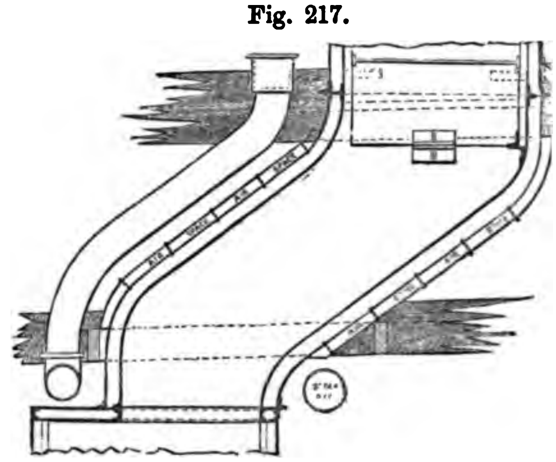
Fig. 216.



UP-TAKE AND AIR CASINGS FITTED IN H.M.S. "RESEAR BY MESSRS. WATT.

up-take has to be prolonged at an inclin angle before connecting with the chimney

Fig. 217.



INCLINED UP-TAKE AND AIR CASINGS BY MESSRS. WA

shown by Fig. 217. In this example mast "stopped the way," and determined t

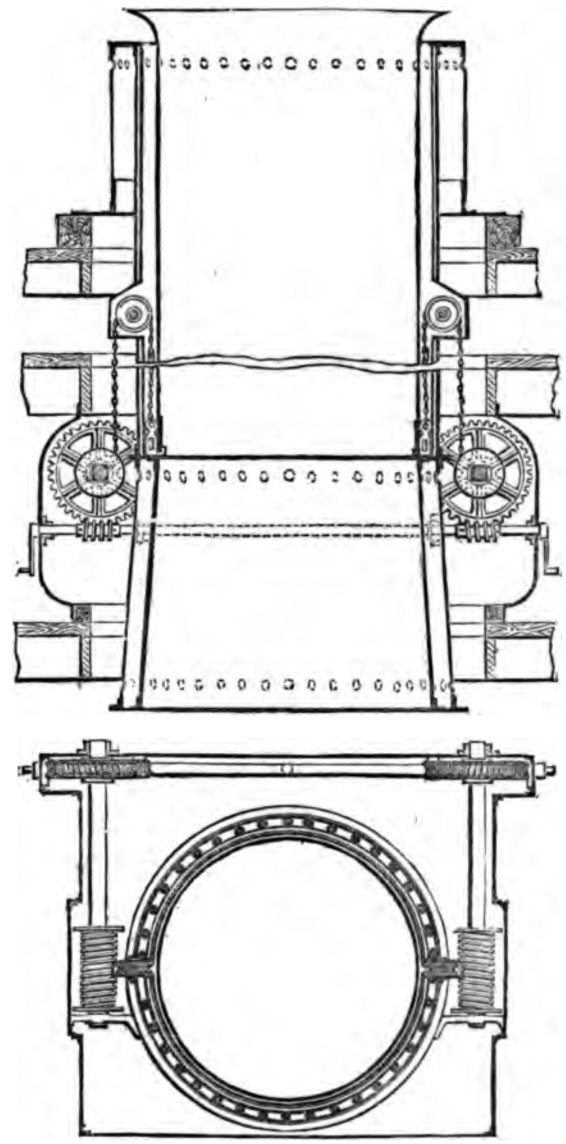
the connection must be beyond the extremity of the boilers. The chimney was situated abaft of the mast, and the up-take extended to make the requisite connection. It is evident, therefore, from these reversed extreme examples, that the situations of the various details determine the requisite form of the up-take.

CHIMNEY HOISTING GEAR.

During naval warfaring engagements it is essential to preserve the portions devoted to the propulsion of the hull. Now the chimney, although forming a strange contrast to the propeller, nevertheless has much to do with the vital effect. It is a common occurrence to lengthen the chimney to attain more draught for the furnaces, and from the same reason it is expedient to prevent damage if possible. Apart, however, from the cause alluded to for the requisition of the gear under notice, it is sometimes preferred to lower the chimney when the ship is under sail or at anchor.

An arrangement of hoisting gear is illustrated by Fig. 218, by the Messrs. Watt, the principle of which is universal for the purpose. The plan illustrates the position of the chain barrel, also the worm wheel and pinion. The elevation shows the position of the chain pulley and barrel between decks, and the handles for manipulation or hand power—which are shown broken to economise space. The action of the gear is thus,—on motion being given to the barrel, the chain is wound around it, and by its connection with the bottom of the chimney, raises it. When the extent of the elevation is attained, a key, or rod, is introduced in the casing, and the chimney rests on the same.

Fig. 218.



MESSRS. WATT'S CHIMNEY HOISTING GEAR.

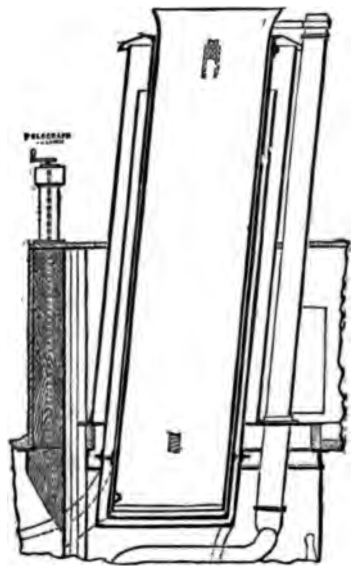
The firm under notice have also fitted an incline l telescopic chimney, shown by Fig. 219 in page 339—being in connection with Fig. 216, in page 338.

TUBES.

The tubes of a marine boiler are of three kinds—plain, stay, and ferruled: examples

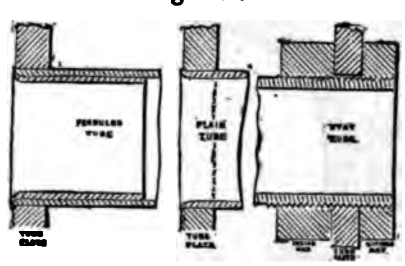
of which are illustrated by Fig. 220. The plain tube is riveted or closed to form

Fig. 219.



CHIMNEY AND AIR CASING FITTED IN H.M.S. "RESEARCH," BY MESSRS. WATT.

Fig. 220.



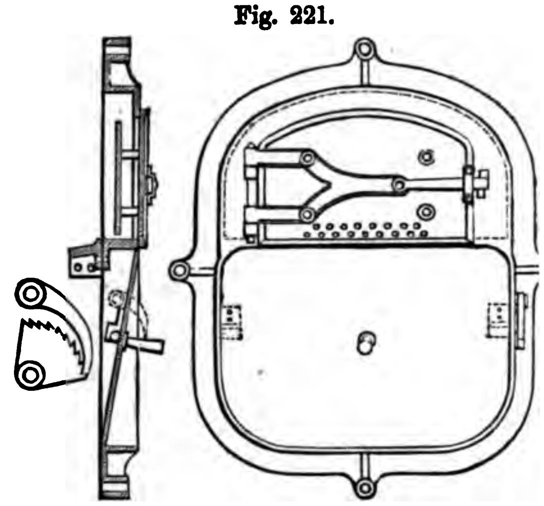
SYSTEMS OF SECURING TUBES.

the connection with the plate; the ferruled example is merely a repetition of the former with a certain length; and the stay tube is secured by nuts, the extremities of the tube being screwed. In the present example the nuts are on each side of the plate, which ensures a perfect joint; but single or outside nuts have been adopted with satisfaction. The position and angle of these tı will be understood by referring to Fig. 12 page 83.

FITTINGS.

First on the list of these details are the and damper doors, with the frame; and illustration, Fig. 221, conveys a correct

Fig. 221.



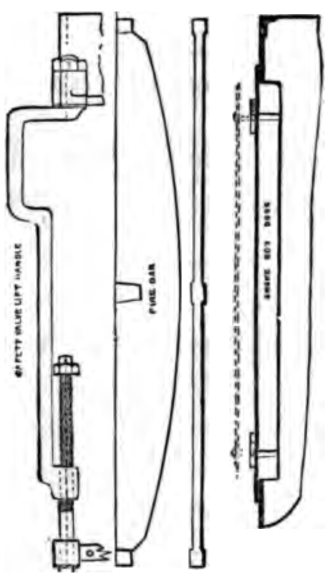
FIRE BOX AND DAMPER DOORS.

as to the general practice. The fire is hung by hinges, and secured when cl by a catch bar and bracket. The door is tected from the heat by a flame plate the inside, as shown in the sectional elevat The damper door is hung by loops res on studs projecting from the insides of frame, and a ratchet quadrant and c retains any position for the door. frame is secured to the boiler plate by st or bolts, passing through the bosses cast the same. It may be added, that a cer amount of clearance or looseness is requ in the connection, to permit the expansio the metal without fracturing the frame securing studs, to say nothing of the leal

ssible. The shapes of the frame and doors ɔ not always alike, which will be seen by uding to Figs. 13 and 14, in pages 93 and 96, ɔse being examples of high and low boilers, ɔ shape of the frame being in accordance th the position of the fire box.

The fire bars are shown by Fig. 222;

Fig. 222.



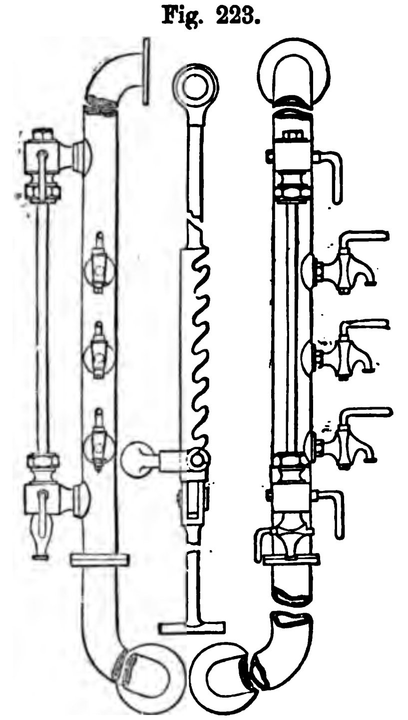
BOILER DETAILS.

o a section of the smoke box door and ɔ safety valve lift handle. The position the bars in the fire box is illustrated Fig. 12, in page 83, where it is seen two s or lengths of bars are adopted. It some-ıes occurs that three sets are used, the dge bar being the shortest.

The smoke box door is generally formed ordinary boiler plates, with a certain space ween them, as illustrated. The dotted por-a shows the air plate outside the door, l the full line depicts a flame plate inside. the first case, the cooling current protects door from the heat inside; and in the second, a plate receives the impact of the heated products. Holes are formed in the bottom side of the door, and thus a current of air passes between the plates. A third arrangement is to have both air and flame plates. The door is hung by hinges at the side, and fastened by lever handles; the stop piece, or inside connection, being a short bar on the handle stud, which, when in contact with the vertical portion between the doors, secure the latter. Illustrations of these details also are shown by Figs. 13 and 15, in pages 93 and 103.

The safety valve lift handle is the ordinary practice—being a crank with a screwed rod to raise and lower the lever under the

Fig. 223.



WATER GAUGE, BRANCH PIPE, GLASS FRAME, GAUGE COCKS, AND SAFETY VALVE LIFT HANDLE.

weights on the valve: another kind of handle is illustrated by Fig. 223. In this case a rod

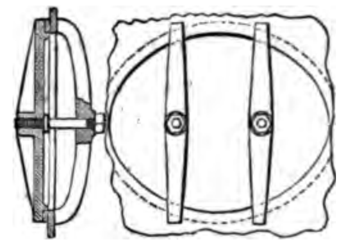
is formed with notches of a sufficient depth to receive a fixed pin, and by pulling the rod downwards, the valve is released, and any position for the lever is retained by the contact of the certain notch and pin. The application of this is shown complete in Plate 35; but by adopting the cranked handle alluded to, in the place of the notched rod, a better effect results. These handles and gear can also be used for opening and closing smoke dampers when introduced, but the air damper, shown by Fig. 221, is the general practice, thus obviating the use of one in the up-take, except when particularly essential. A third means of lifting the valve and weight is illustrated by Fig. 23, in page 144.

The indication of the water in the boiler is the most important acquisition in the marine type: this is usually attained by a glass tube and gauge cocks, as illustrated also by Fig. 223. As the smoke box extends the entire length of the tube openings, the gauge frame cannot be connected direct to the boiler front, thus a branch pipe of suitable form and dimensions is used as illustrated. The lower flange is secured below the smoke box door and the upper above it; the pipe projects from the front, sufficiently to open the door in question. The frame and cocks are secured on the pipe, and thus a neat arrangement is effected.

To inspect, cleanse, and repair the interior of the boiler, doors are requisite, and these and the mode of securing them are depicted by Fig. 224. The hole cut in the plate is surrounded at the edge by a band—which should never be omitted. The joint is made at the inside, and the connection by the bolts, nuts, and brace or cross bars at the outside.

In connection with the lift handles allud to are the safety valves shown by Fig. 22 below. This is an arrangement on the corre

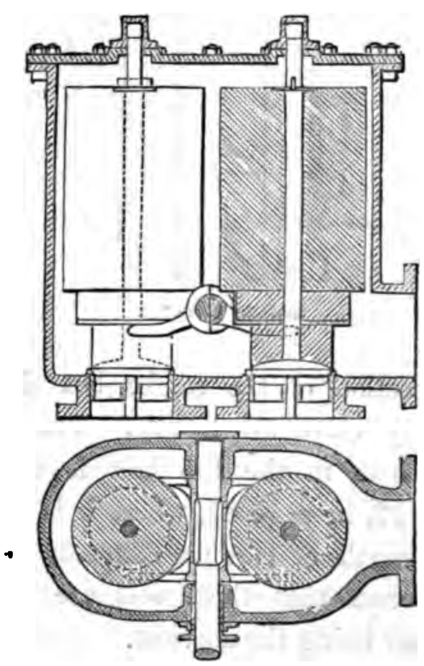
Fig. 224.



MUD HOLE OR MAN HOLE DOOR.

principle, because the weights are direct on th valve, and secure from being tampered with b

Fig. 225.



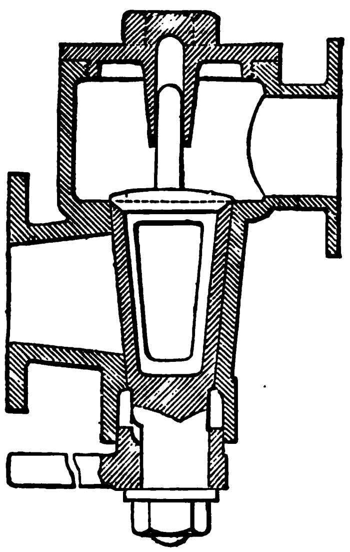
DIRECT LOADED SAFETY VALVES.

ignorant and foolish individuals. The weigh are lifted to release the valve by levers, and tl valve is guided below the seat, and above tl load. Independently of the nuts securing tl cover, padlocks can be connected, or an ordinai

box lock, if preferred. This example is the universal practice amongst all the leading engineers. The casing can be secured to the front or top of the boiler, according to the head space available, better to be conceived by reverting to Figs. 29, 34, and 36—pages 163, 170, and 174.

Next on the list is the feed cock and non-return valve, both contained in one casing, as shown by Fig. 226. The plug is

Fig. 226.



FEED COCK AND NON-RETURN VALVE.

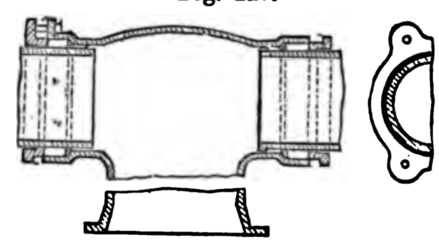
secured at the small end; the valve seat is on the opposite extremity, being guided by a spindle and tube above.

The surface blow off and blow out cocks are as the example illustrated by Fig. 65, in page 235. In some cases, however, these cocks are as the Fig. 226, minus the valve; the casing having a flange, directly beyond the plug, to attach to the boiler.

The operation of blowing off is often assisted by a scum or brine trough, as shown by Fig. 14, in page 96, in connection with the cock, and thus the collection on the surface of the water is more readily conveyed to the point of egress or escape.

The communication of the boilers and engines is effected by steam pipes, and to permit the natural expansion and contraction of the metal, stuffing boxes are often used, more particularly with branch pipes, as shown by Fig. 227. With straight pipes the metal is

Fig. 227.



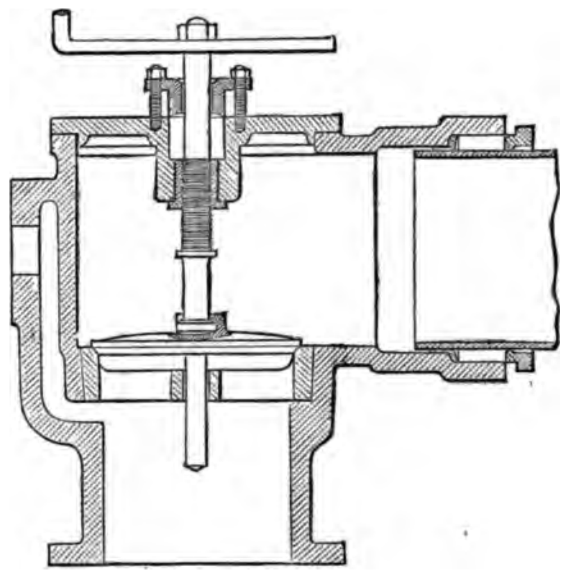
STEAM PIPE BRANCH PIECE.

often curved outwards, and thus a spring is attained. The application and arrangement of this mode is shown by Plate 33, and a flanged connection by Plate 2.

Combined with the steam pipe is the stop valve, and a universal example is illustrated by Fig. 228, page 344. This is a disc valve raised and lowered by a screwed rod above it, and a cranked handle for manipulation. The passage at the side is for a supplementary small valve when required for separate purposes; thus obviating a second joint and connection with the boiler plate. The discharge pipe is fitted in a stuffing box, while in some instances a flanged connection is only requisite. The arrangement of the pipes and valves depends on the number and relative positions of the boilers and cylinders in the hull; consequently no arrangement can be decided on for general practice. Examples of the requirements of stop valves are

illustrated by Figs. 13, 14, 28, 29, 31, 32, 34, and 36, in pages 93, 96, 161, 163, 164, 167, 170, and 174.

Fig. 228.

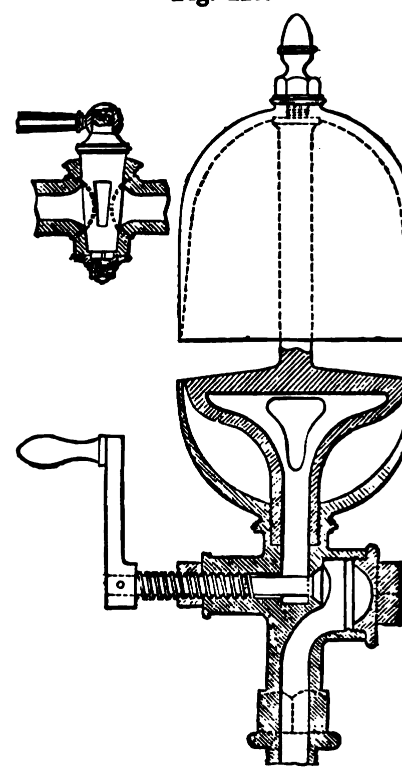


STEAM STOP VALVE.

The adoption of steam as an alarm as well as a motive power was a happy idea, and the mechanical arrangement to produce the result a feat of ingenuity. Steam whistles are now as common as requisite, and the variation in their form depends on the note to be sounded. The example shown by Fig. 229 is an ordinary type, with a stop valve and screwed spindle. It is preferred, however, to adopt a plug —shown in section at the side of the bell—in some instances to regulate and stop the escape of the steam. The action of the steam is to rush through the annular space and strike the bell above. The contact of the vapour and the metal causes the latter to emit a continual note of music, often more expressive than agreeable.

The pipes conducting the steam from the safety valves are arranged according to the relative positions of the casings and chimn and therefore with them also. Circumstan alter examples of arrangement.

Fig. 229.



STEAM WHISTLE.

The arrangement of the pipes and fittir of marine boilers is represented in Pla 23, 24, and 30; and for steam launches, Plates 29, 31, 32, and 34, which enables practical conclusion obvious without furtl description.

CLOTHING STEAM PIPES, AND LAGGING CYLINDERS AND BOILERS.

To preserve the temperature of the steam much as practical during the traverse fr the boilers to the cylinders, the pipes ı clothed. The method of this operation is wind around the pipe a portion of felt,

other non-conductive material, which is secured by wire or cord; stout canvas is sewn around the felt, and thus a plain exterior is attained—the appearance and effect being further improved by one or two thick coats of paint on the canvas.

Cylinders are lagged by felt and cross strips of wood, the finish being uniform pieces of wood secured longitudinally by wrought iron or brass bands, as shown in Plates 3 and 25.

Lagging a boiler consists of covering all the available exterior surfaces—back, ends, and top—with felt and wood. The mode often adopted is, to insert a certain number of studs in the shell of the boiler—in some instances in the ends of the stays—the felt is laid on, and secured by battens of wood, the latter being held by the studs. The recesses between the battens are filled with felt, and planking secured to the battens completes the operation. The top of the boiler is covered with lead outside the battens, instead of wood, as the ends and back; this is to prevent any water that may accumulate at that place from settling and corroding the plating.

Steam pipes are sometimes lagged also, similar to the cylinders, while in some instances cement has been laid on thick in the place of any other covering.

COAL BUNKERS.

The form of these storing rooms depends on the size of the hull, relative to the space occupied by the engines and boilers, and the quantity of fuel to be stored. The material and mode of construction consists of angle iron as supports, and thin plate or sheet iron to form the sides and ends. The structure is stayed also to prevent deflection or bulging The dimensions of the material generally used at present are—thickness of top plates $\frac{1}{8}$ of an inch for a certain depth, below this $\frac{3}{16}$-inch plates are used; the angle iron used in the corners is $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{4}$ inch, the stays $2 \times 2 \times \frac{5}{16}$ inch, and the pitch of the latter 3 feet. It is essential to use temperature tubes amongst the fuel; each tube being from 3 to 6 inches in diameter, into which a thermometer is lowered to indicate the temperature. The number of tubes are one per 30 tons of fuel stored in bunkers containing above 200 tons.

CHAPTER IX.

THE PRINCIPLES OF THE MARINE ENGINE.

To set aside the principles of any subject is similar to building a structure without knowing the status of the foundation; it is also apparent that the effect of any cause is due to its origin; the main question, at present, however, is the best method of arriving at the truth of certain results, and at the same time to develop the means for further improvement.

The marine engine, as it is at present, consists of a sliding action imparting motion to a pin which revolves around the axis of suspension or support, and therefore our first subject refers to the actual difference in the speed of the piston and crank pin at equal positions on the centre line of motion.

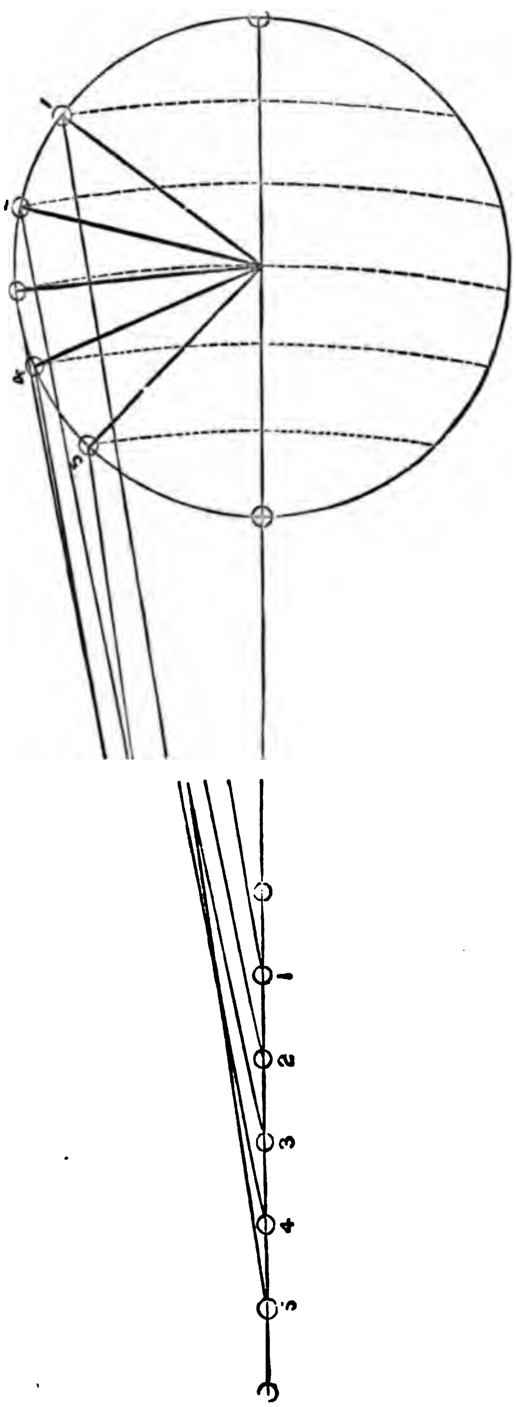
Variation in the Speed of the Piston and Crank Pin.—The diagram, Fig. 230—on page 247—represents the paths of the cross-head or piston, and crank pin; the stroke is 3 ft. 6 in., and the length of the connecting-rod between centres is 9 ft. 6 in., being less than three times the stroke of the piston. The plane line of motion is divided equally into six parts, thus producing five points of cut-off, or grades of expansion. Now it is obvious that the piston, on arriving at 1, caused the crank pin to be at 1 on its path, but the distance passed through is in the proportion of about 1 to 3. On the piston arriving at 2, the crank pin has advanced to the same number on its circuit, but the length passed is about as 7 is to 9; from 2 to 3, and 3 to 4, the passages in each case are more equal; but from 5, to the completion of the stroke, an inequality of importance again occurs in the proportion of 7 to 12·35. With a shorter connecting-rod the variation between all the points of motion on the circular path will be effected in due proportion to the radii of the dotted curves intersecting with the circle in question.

From this diagram, or any other of the same principle, the variation in the speeds of the piston and crank pin can be easily known, also the time of admission of the steam in proportion to the amount of duty effected. For example, if the piston is moved forth and back from the end of the stroke to 5, sixty times in each direction in a minute, the distance traversed $= 14 \times 60 = 840$ in.; but the crank pin's passage $= 42 \times 60 = 2520$ in., and thus the speed of the piston for one-sixth of the stroke is one-third of the crank pin's velocity. With this inequality, however, there is a certain gain, for the steam admitted is more powerful before the cut-off than after, or during expansion, and therefore when the greatest effect is requisite it occurs.

Delineation of the Path of the Crank Pin.—The motion imparted to the slide valve is generally derived from two principles of action—vibratory and rotary. Now, when the former is the prime mover, the speed of the valve is the same throughout the stroke, or rather, if the motion is imparted by the piston, the motion of it and the valve would be equal. Rotary motion is more often adopted than any other for the transmission of power and action, and to the present day the small cranks and circular eccentrics are the prevailing means employed to impart the motion required for the slide valve. The relative speeds of the crank and the eccentric are proportionately the same in theory and practice. The length of the connecting-rod in all examples of rotary motion regulates the inequality of the speed of the sliding body. It must be remembered, however, that a reciprocity of motion can be attained by a correct proportion of the details.

Before proceeding further with the definition of the better arrangement, it will not be out of place to define the relation of the path of the crank to the sliding motion imparted to the valve, also the controlment of the steam. It is doubtless universally known that, virtually, the crank path is divided into four distinct parts, also that for the eccentric. The proportions of these divisions are practically regulated by the grade of expansion agreed on to be maintained. Fig. 231 represents a crank pin's path with the chords indicating the divisional points, the

Fig. 230.

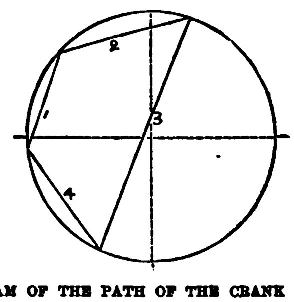


Scale ¾ inch = 1 foot.

...'S DIAGRAM OF THE PATHS OF THE PISTON AND CRANK PIN.

...tive proportion is, therefore, readily understood. ...1 is the chord of supply, No. 2 that for expansion, 3 relates to exhaustion, and No. 4 represents neutrality, or that portion of the stroke of the piston when the port on the exhaust side is covered, often termed compression; consequently, the piston for a period is

Fig. 231.



BURGH'S DIAGRAM OF THE PATH OF THE CRANK PIN IN RELATION TO THE SLIDE VALVE'S ACTION.

devoid of pressure or vacuum. The length of the chord 1 is due to two causes, which are the grade of expansion and the length of the connecting-rod. It will be noticed that the chord at the plane line intersects with the circle slightly below the same. This last intersection is the angle that the crank assumes when the slide valve commences to open the port, and the vertical distance from the intersection to the plane line is due to the lead required. The upper point of intersection, as before explained, is subject to the curve assumed by the connecting-rod from a point, or distance, on the plane line to the circle of the crank path. The length of chord No. 2 is regulated by the inside and outside laps of the slide valve. The expansion of the steam is now in full operation, and is released by the opening of the port on the exhaust side, hence the intersection of the chord at No. 3. The length of the last-mentioned chord is more than any other, due to the traverse of the valve, or the time occupied in opening and closing the port for exhaustion. Now in the case of an increase of supply steam, the time for expansion and exhaustion would be lessened in proportion, it being remembered that the circle described by the crank pin cannot be increased or decreased for a given length of stroke of piston. The circle, as before stated, is divided into four divisions, and the alteration in the grade of expansion or length of connecting-rod affects the whole proportionately. The concluding chord, No. 4, represents neutrality, or, as before stated, that portion of the stroke of the piston where the vacuum and steam is cut off for a given period, commonly known as compression. It will be remembered that the chord of expansion is due to the outside and inside laps, *i.e.*, when the valve is at the edge of the supply side of the port, the valve has to

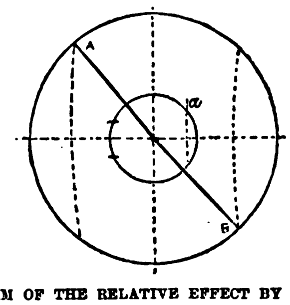
travel forth until the inside edge permits exhaustion or destroys expansion. The valve is now at half stroke, plus inside lap; exhaustion of steam, therefore, must ensue until the valve is in the same position, but travelling in a reverse direction. The position of the valve when terminating exhaustion will be half stroke, minus inside lap. It can thus be clearly understood that the length of the chords for expansion, and that for compression, are equal in the example given; it may also be added that any variation in these two chords will depend on various causes, such as unequal laps and leads, &c , &c., but with certain arrangements both are equal. The fact of the compression being the same as the expansion, is of no vital importance. It is certain there would be a gain in maintaining expansion longer, and exhausting, till the supply commenced, and thus dispensing with compression; but the present motion of the slide valve would have to be altered, as an extreme unequal action would have to be attained.

The proportions of the path of the crank pin can thus be clearly understood. As at present arranged—compression, supply, and expansion, form one portion of the circle, and the remainder is occupied by exhaustion, which is the most. The diagram given relates to one revolution of the crank only, it being well known that all in principle are alike. There is, however, a slight variation in practice, due to the versed sines of the rods, and inequality of the length of arcs passed through, in proportion to the propelling or sliding motion attained.

The arrangement of the slide valves, in relation to that of the connecting-rod, should be considered as to the attainment of equal action. The versed sines of the chords of the arc passed through by the crank pin, at each angle of the stroke, for a given supply of steam, are unequal, also the versed sines of the eccentric, when the slide is at the same side of the crank as the connecting-rod; but this can be counteracted to a certain extent by short eccentric rods, and levers, reverse in action, and arranged to compensate for the inequality of the speed of the circular and sliding motions, the attainment of which being of great importance.

Fig. 232 represents the circle described by the crank pin for a given stroke of piston. The horizontal line is presumed to be the centre of the engine, and the larger dotted arcs represent an even grade of expansion, on each side of the piston or at each stroke. Now it will be seen that on the crank—represented by the thick lines—moving from the plane to the intersection at A, a given length of stroke of piston is produced, due, of course, to the radius of the larger dotted arc. On the crank reaching to the intersection at B, the grade of expansion is reversed in action, but the same distance

Fig. 232.



BURGH S DIAGRAM OF THE RELATIVE EFFECT BY THE POSITIONS OF THE SLIDE VALVE AND CONNECTING ROD.

from the end of the stroke retained. Now, the difference in the lengths of the arcs passed through are due to the length of the dotted arcs, the radii of which are the connecting-rod. The smaller circle *a* indicates the travel of the valve, or the path of the centre of formation of the eccentric. The dotted arc indicates the distance the valve must be from the edge of the port, when the piston is at full stroke, hence the angle or advance of the eccentric to that of the crank when on the horizontal line. When the crank is at A, the eccentric has passed through an arc equal in length proportionately. The two dots on the circle opposite the dotted arc indicate the angles of the eccentric when the crank is at the plane line and at B. Now it will be readily seen that the space between the dots is less than that of the intersections opposite; also the versed sine reduced. It is obvious that an unequal grade of expansion must ensue on the side of the circle at B., *i.e.*, if the laps of the valve are equal. It must also be remembered that, to increase the lap and retain the previous stroke of the valve is to destroy the lead; hence, the slight variation in the time for supply steam at each stroke of the piston may be said to be endured rather than introduce a worse evil.

It is not here intimated that perfection of mechanism should not be attained, but rather to delineate and describe that most universal. The diagram now under question, illustrates the principle of the action of the crank and eccentric with the valve situated on the same side of the crank shaft as the connecting-rod, direct action in each case being maintained. It will be seen that the arcs are all struck by the radii on the same side of the perpendicular line, hence the variation above alluded to. Now in order to counteract, or rather ob-

viate the imperfections now under notice, the lengths of the chords indicating the partial stroke of the valve must be equal on each side of the perpendicular line.

Fig. 233.

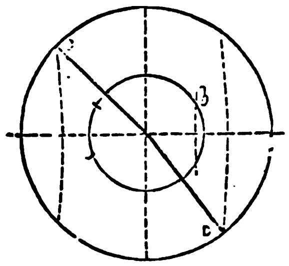


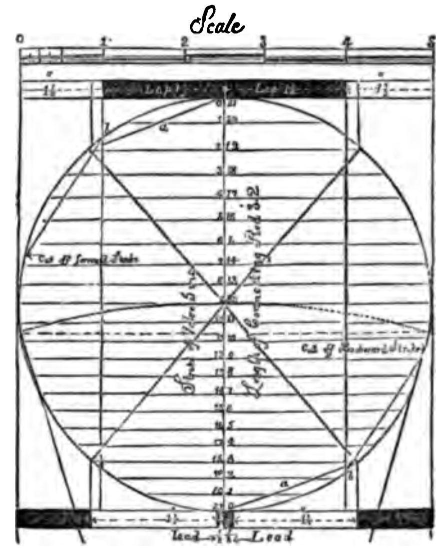
DIAGRAM OF THE RELATIVE EFFECT BY THE POSITIONS OF THE SLIDE VALVE AND CONNECTING ROD.

Fig. 233 represents a crank pin path, and that of the eccentric, to cut off at the same grade of expansion as Fig. 232. In the present case the arcs passed through are reversed to that of the former, the piston presumed to be moving in the same direction, but the situation of the connecting-rod opposite to that of the slide valve. It will be noticed that the diameters of the eccentric paths are unequal; this inequality is due to the variation in the arc of the crank pin's passage during the forward and backward grades of expansion. When the crank has moved from the centre line to C, the eccentric has passed through an equal arc, and the same relative motion occurs from the horizontal line to D. Now the length of the intersection at *b* can be seen, in proportion to that between the dots in the circle opposite. Suffice it to say, the nearer these two intersections agree in length or space between the same, the less variation will ensue in the grades of expansion at each double stroke of the piston. It will be remembered that the position of the pistons in Figs. 232 and 233 are presumed to be alike; also the direction of the movement. It may as well be added that on reversing the action of the piston, the action of the valve will be affected.

Geometrical Demonstrations of the Slide Valve.—The subject now entered on has been well digested by many authors, both English and foreign. Of the latter the best authority is Dr. Gustav Zeuner, a German gentleman: his researches and expositions pertain, however, more to the locomotive than the marine type; but, in the main, his views are correct for either, under relative circumstances. The views of our American friends are represented in a work termed the "Cadet Engineer." At home, Messrs. Watt, Professor Rankine, Mr. D. K. Clarke, C.E., and Mr. M'F. Gray, claim attention for their productions; and we have also done something to solve the problem.

Messrs. Watt's mode of treating the matter under notice is illustrated by Fig. 234, which represents the

Fig. 234.



MESSRS. WATT'S MODE OF PRODUCING THE POINTS OF CUT-OFF BY A KNOWN LAP.

path of a crank pin 21 inches in diameter; length of connecting-rod between centres 3 feet 2 inches, and the stroke of the slide value 5 inches. The method of utilizing the diagram is thus: "Divide the path of the cross-head pin into inches; with the connecting-rod's length as a radius, and each inch as centres, describe arcs, intersecting with the path of the crank pin; join the intersections by chords or lines, as depicted, and the result is a correct illustration of the relative positions or progress of the crank pin and piston. Next draw vertical lines parallel to the centre line as tangents, and above the circle, between these tangents, form a scale of as many equal divisions as the number of inches in the stroke of the valve, which is virtually assuming that the stroke of the piston is that of the valve. Now, having previously settled the outside lap of the slide valve, next draw two tangents parallel with the centre horizontal line, and on the top tangent on each side of the vertical centre line, set off the lap according to the scale: the lap being 1½ inches actually, 1½ divisions in the scale is the length required, each main division in the scale being virtually inches. Draw vertical lines from the laps cutting the circle, and prolong them to the bottom tangent. On each side of the centre line

set off the lead of the slide valve, at the same scale as the lap, on the lower tangent, and from the laps set off the leads also. From these last points draw lines parallel with the vertical centre line, to intersect with the circle above and below; connect these intersections by angles crossing each other at the centre of the circle. Then with the chord a as a radius, and b as a centre, describe an arc on the circle, in reverse localities, and each last intersection is the centre of the crank pin, when the slide valve has closed the supply ports, during the forward and backward strokes of the piston. The curve seen intersecting with the circle's centre is described by the connecting-rod when the piston is at half stroke; and, by a similar process, the position of the piston can be ascertained at the points of cut-off, which in this case is a mean of 12·82 inches from the commencement of the stroke. As the top and bottom tangents represent the steam and vacuum sides of the slide-valve, the position of the latter, and angle of the eccentric, are also obvious.

Professor Rankine, when treating of the Slide-valve Gearing, in his work of "Rules and Tables," states:—

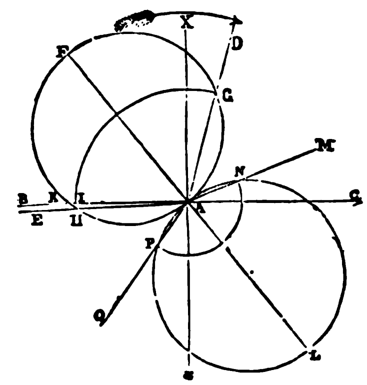
By the angular advance of the eccentric is to be understood the angle at which the eccentric radius stands in advance of that position which would bring the slide valve to mid-stroke when the crank is at its dead points.

Rule I.—Given, the positions of the crank at the instants of admission and cut-off; to find the proper angular advance of the eccentric, and the proportion of the lap on the induction-side to the half-travel of the slide.*

In Fig. 235 let A B and A C be the positions of the crank at the beginning and end of the forward stroke; let the arrow show the direction of rotation; let X x be perpendicular to B C; let A D be the position of the crank at the instant of cut-off, and A E its position at the instant of admission. Draw A F, bisecting the angle E A D; A F will represent the position of the crank at the instant when the slide is at the forward end of its stroke; and F A X will be the angular advance of the eccentric.

Lay off the distance A F to represent the half-travel; and on A F as a diameter describe the circle A H F G, cutting A D in G and A E in H; then $\frac{AG}{AF} = \frac{AH}{AF}$ will be the required ratio of lap at the induction-side to half-travel; and A G = A H will represent that lap, on the same scale on which A F represents the half-travel.

Fig. 235.



DR. ZEUNER'S GEOMETRICAL DIAGRAM OF THE SLIDE VALVE AND CRANK.

On the same scale, I K represents the width of opening of the valve at the beginning of the stroke, sometimes called the "lead of the slide." Strictly speaking, this is the lead of the induction-edge of the slide only; the lead of the centre of the slide being A K; that is, its distance from its middle position at the beginning of the forward stroke.

Given the data and results of the preceding rule, and the position A M, of the crank at the instant of release, to find the ratio of lap on the eduction-side to half-travel, and the position of the crank when cushioning begins. Produce F A to L, making A L = A F; on A L as a diameter draw a circle, cutting A M in N; then $\frac{AN}{AL}$ will be the required ratio of lap at eduction-side to half-travel.

About A draw the circular arc N P, cutting the circle A L again in P; join A P; then A P will be the required position of the crank at the instant when cushioning begins.

Rule III.—Given, the data and results of Rule I., and the position, A Q, of the crank at the instant of cushioning; to find the ratio of lap at the eduction-side to half-travel, and the position of the crank at the instant of release—produce F A as before; on A L = F A as a diameter draw a circle cutting A Q in P: $\frac{AP}{AL}$ will be the required ratio of lap at the eduction-side to half-travel.

About A draw the circular arc P N, cutting the circle

* The method used in this and the following rules is that of Professor Dr. Zeuner of the Swiss Federal Polytechnic School at Zürich, published in his treatise on Slide Valve Gearing, entitled *Die Schiebersteuerungen.*

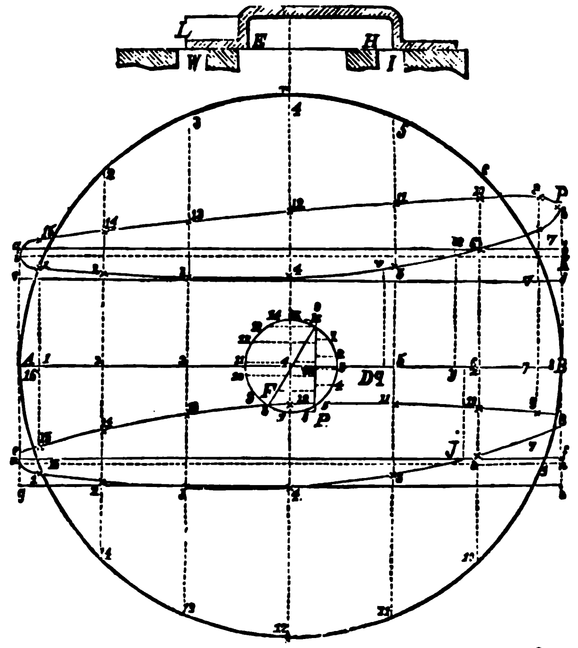
A L again in N; join A N: A N will be the position of the crank at the instant of release.

Given, the angular advance of the eccentric, the half-travel of the slide, and the lap at both sides; to find the positions of the crank at the instants of admission, cut-off, release, and cushioning. Draw the straight lines B A C and X A x perpendicular to each other, and take B and C to represent the dead points. Let the arrow denote the direction of rotation. Draw F A L, making the angle F A X = the angular advance of the eccentric; and make A F = A L = half-travel. On A F and A L as diameters, draw circles. About A, with a radius equal to the lap at the induction-side, draw an arc cutting the circle on A F in H and G; also, with a radius equal to the lap at the eduction-side, draw an arc cutting the circle on A L in N and P. Draw the straight lines A H E, A G D, A N M, A P Q. These will represent respectively the positions of the crank at the instants of admission, cut-off, release, and cushioning.

The information alluded to in the "Cadet Engineer" is described and illustrated as follows:—

Now, if we wish the port to be closed before the termination of the stroke, we make the face of the valve longer, or put on lap. In this case the throw of the valve must be increased by an amount equal to twice the lap. But if excessive lap be put on, it is evident that the travel will be so much increased as to permit the steam to exhaust at an early part of the stroke. To obviate this, we must put lap on the exhaust side of the valve. This has a bad effect in causing the exhaust valve to close too soon. This will be seen clearly in the illustration of the geometrical action of the slide valve, Fig. 236. Let A B equal the length of stroke of the engine drawn to any scale. We will give the valve an amount of lap on the steam side equal to half the breadth of the steam port. The travel of the valve will then be equal to three times the breadth of the steam port. On A B, as a diameter, describe a circle which will represent the path described by the centre of the crank pin, while the piston is travelling twice the distance A B. Divide this circle into any number of equal parts, and draw perpendiculars to A B from every point of division. We shall thus determine the position of the piston corresponding to that of the crank at various points. With the same centre t, as that of the circle A r B s, describe a circle C o D p, having the travel of the valve for its diameter. This will represent the path of the centre of the eccentric during a revolution of the crank. When the crank is on the centre, the line connecting the centre of the crank pin and the centre of the shaft will be A t; so that if the valve had

Fig. 236.



AMERICAN GEOMETRICAL DIAGRAM OF THE SLIDE VALVE.*

neither lap nor lead, the line connecting the centre of the eccentric and the centre of the shaft should take the direction t r, perpendicular to A t. But in the present case, when we have both lap and lead, we make t u equal to the sum of the lap and lead, and through u draw a line parallel to t r. Connect the point o where this line cuts the circle with the centre, and o t will be the proper position for the line connecting the centre of the eccentric and the centre of the shaft, when the crank is on one centre. When the crank is on the other centre, this line will appear at t F. Divide the circle C o D p into the same number of equal parts as we divided the circle A r B s, and draw perpendiculars to o p from every point of division. The lengths of these perpendiculars show the distances travelled by the valve at various points. Now, let A B represent the centre of the exhaust port. Then draw a b and c d to represent the width of one steam port, and e f and g h for the other. Make a i equal to L W, the steam lead, and draw a line i k parallel to a b. This is the line to which all the measurements must be referred, since the valve commences to move from this position. Thus, when the crank has moved the distance A 1, the centre of the eccentric has moved

* The curve from a to P, should be more as that below, e to B.

the distance o 1, and the perpendicular distance of this point 1 from $o\ p$ will be the distance the valve has travelled. Lay off this distance below the line $i\ k$ on the first perpendicular, and the point so determined will represent one position of the valve. Similarly, when the crank has travelled the distance A 2, the centre of the eccentric has passed over o 2, and the valve has travelled the perpendicular distance between 2 and $o\ p$. Lay off this distance on the second perpendicular below $i\ k$, and we determine the position of the valve at another point of the stroke. Find the position of the valve in this manner at every point of division, and through the points so found draw a curve which will represent all the positions of the valve during one revolution of the crank. It must be observed, in laying off perpendicular distances from $o\ p$, that all points to the right of $o\ p$ are positive, and are laid off below $i\ k$, while all points to the left of $o\ p$ are negative, and must be laid off above $i\ k$. We have not yet determined the amount of exhaust lap, but this can readily be fixed, now that the motion of the valve is represented. When the piston has made one stroke, the crank is at B and the valve is at F. Now, if the face of the valve was only $1\frac{1}{2}$ times the width of the port, the whole port would be open for the steam to exhaust. So we must put lap on the exhaust side of the valve, and we put on enough to have the exhaust lead equal to H I. This gives us the width W E, of the valve face.

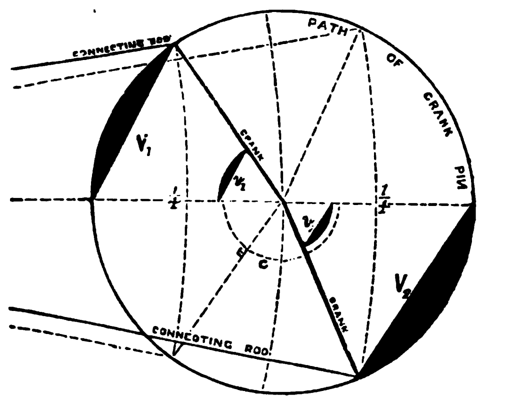
From the curve $a\ i$ P, we can readily find the position of the valve, corresponding to any position of the piston. Thus, when the piston has travelled the distance A q, the valve is at v, and the distance the port is open is equal to the perpendicular distance between v and the top of the port $a\ b$. At x, where the curve cuts $a\ b$, the port is closed, and the steam expands during the remainder of the stroke from y to B.

We can readily lay down the curve described by the lower extremity of the valve. The distance $e\ m$ between $e\ f$, the top of the port, and $m\ n$, is the exhaust lead, and we have only to transfer the distances of the various points in the first curve from $i\ k$ to their respective distances from $m\ n$, on the same perpendicular on which they were first laid down. By this curve we see that the exhaust opening is closed at j, when the piston has travelled the distance A z.

During some portion of the year 1865, the Author wrote a series of articles over and under his name, which were inserted in the pages of the scientific journal, the "Engineer." He there stated, when alluding to the lap of the valve, thus:—"The rules given by many authors for defining by calculation the amount of lap due to the grade—expansion—and stroke of the piston are useful, and no doubt are of value to a certain extent. Rules, however, if complicated, cause mistrust, and in some instances create confusion of ideas or perception where none should exist. Now, in most if not in all these rules, the travel of the valve is mentioned, which, in fact, is the most important feature in the formulæ. This is, of course, correct, when the width of the ports—or openings—are not known, and the travel of the valve is assumed. Now, as the ports are generally determined before the valve and gear is proportioned, the rules should therefore be *seriatim*." It is, of course, apparent from this that the grade of expansion, being settled, and the width of the opening caused by the slide valve known, the *outside lap should define the stroke of the valve*. On returning to page 347, and consulting Fig. 230, it will be seen that the angles of the crank, at the points 1 and 5, are unequal, although the positions of the piston from each end of the stroke are at equal distances. Now, as the eccentric's point of formation, and the centre of the crank pin both rotate on the same axis, and the angles of each, in relation to each other, are unalterable, *the paths of the crank pin and eccentric are alike in principle of action*. Neither must it be forgotten that all *versed sines* bear a relative proportion to the radii or diameters of the respective circles. The question may be raised, What *can* the versed sine have to do with the lap of the valve? The best answer will be a reference again to Fig. 230 —page 347—it will be seen that, as the *arcs* passed through by the crank pin, from the plane line to 1, and from the plane line to 5, are *unequal* in length, the versed sines must be similarly effected, because the radius is unalterable. Now then for the application of this fact in practice. The illustration, Fig. 237—page 353—represents the path of a crank pin 2 feet in diameter, and the length of the connecting-rod between centres is 5 feet, or 5 cranks; the grade of expansion $\frac{1}{2}$ from the commencement of the stroke. When the crank pin has passed through the *arc* which produces the chord V_1, the crank is at a given angle, due to the length of the connecting-rod, and the piston has advanced 6 inches. On the completion of the stroke the crank descends, and when the chord V_2 has been produced, the piston has returned 6 inches, but the angle of the crank more obtuse than the former alluded to. It may be added that the *length* of the connecting-rod, in proportion to

stroke of this piston, determines this inequality in all ı. Now, the versed sines, V_1, and V_2, are in pro- ion to the angles of the cranks, or the lengths of the

Fig. 237.



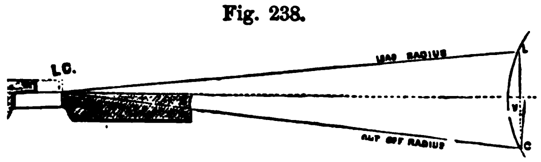
Scale 1 *inch* = 1 *foot.*

ıGH'S MODE OF DETERMINING THE LAP OF THE SLIDE VALVE, POINTS OF CUT-OFF, AND POSITIONS OF THE ECCENTRICS.

ds of supply, to which attention has been directed Fig. 231, in page 347. The ratio of the versed V_1 is $\frac{1}{18}$th of the diameter of the crank-pin's path, V_2 $\frac{1}{12}$th of the same.

he proportions of the steam ports in the cylinder t next be noticed: Width of opening caused by ı ¾ inch, lead ¼ inch: so that when the piston is at . end of its path or the crank on the plane line on ır side, the slide valve has opened the port ¼ inch, therefore ½ inch more travel is the completion of stroke of the valve. To return to the diagram, 237, is now necessary: it will be seen that near centre of the circle are two lesser shaded portions, nd v_2, corresponding in form and proportion with ınd V_2 respectively; these are termed the versed ı of the eccentric, while the larger are known he versed sines of the crank. The author has ıd in his work on the Slide Valve, that the true ed sine of the eccentric arc of supply steam equals *h of the opening caused by the slide valve minus the* . The certainty of this conclusion is evident from

Fig. 238.



ıAM OF BURGH'S MODE OF PRODUCING THE "VERSED SINE" OF THE ECCENTRIC.

Fig. 238. This represents a steam port and end of the lap portion of the valve at full travel; the dotted line L is the position of the valve for the lead, and at C when it is closed. It has been clearly proved that the lengths of the connecting-rods affect the travel of the piston, and it is therefore obvious that the lengths of the eccentric rods affect the travel of the valve. The lines L L and C C are the length of the distance between the centres of the crank-shaft and block-pin in the link; the centre of the pin being virtually the ends of the slide valve. Then with L at the dotted line as a centre, describe the arc cutting the plane line and the eccentric circle at L, and with C as a centre describe an arc cutting the circle at C; the distance between the points of intersection on the plane line is the amount of the lead, or as the space between the dotted lines L C. Join L C on the circle by a chord or a straight line, and the intersection at V is the *versed sine of the eccentric.*

Now if the versed sines of the crank equal respectively $\frac{1}{18}$th and $\frac{1}{12}$th of the stroke of the piston, the mean ratio will be $\frac{1}{14}$th. Then if the versed sine of the eccentric equals ½ inch, $\cdot 5 \times 14 = 7$ inches or the stroke of the slide valve, as shown in dotted lines—E C —inside and outside the shaded portions v_1 and v_2 in Fig. 237. Alluding further to this diagram, it is obvious by its use that the lap of the slide can be known; for *any* grade of cut-off, *any* lengths of eccentric and connecting rods, *any* amount of lead, *any* width of opening caused by the slide, *any* length of stroke for the piston, and for *any* deviation; in general practice by the following simple rule: *divide the radius of the crank circle by the versed sine of the crank; multiply the quotient by the versed sine of the eccentric; the product, minus the width of the opening*, equals the outside lap.

For example, the travel of the valve in the diagram is stated to be 7 inches, and the versed sine of the crank $\frac{1}{14}$th of the stroke of the piston; then $\frac{24}{14} = 1 \cdot 7142$, or

$$\frac{12}{7} = 1 \cdot 7142, \; \frac{12}{1 \cdot 7142} = 7 \cdot 0 \times \cdot 5 = 3 \cdot 5 - \cdot 75 = 2 \cdot 75$$

the outside lap of the slide valve.

The positions or angles of the eccentrics outside the crank's centre line are known by the chord of the eccentric's versed sine, intersecting with the eccentric circle.

Width of Ports and Bars in the Slide Valve and Cylinder.—Having settled the question of lap and travel of the valve, it is next requisite to know the arrangements of the ports and the solid spaces between them, termed bars,—which are represented fully

in pages 284 and 285—and next, the proportions of the component parts, taking the form of valves illustrated as examples.

Area of Opening caused by the Slide Valve = horse-power nominal × 1 to ·75 square inch (area of supply port = area of opening × 2).

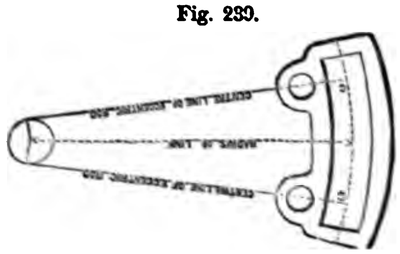
Width of Exhaust Space in Valve = width of two ports supply × 1·5 or 2 + half travel of valve + width of small bar in cylinder − inside lap.

Width of Exhaust Port in Cylinder = width of small bars in cylinder − inside laps of valve, deducted from the width of exhaust space in valve.

Width of large Bar in Cylinder = travel of valve + width of small bar in valve.*

Link Motion—Geometrical Demonstrations.—The main question with this detail is the radius; and the illustration, Fig. 239, shows the general practice of

Fig. 239.



GENERAL MODE OF DETERMINING FORM OF LINK.

determining the same and form of the link, as follows. Describe the circle denoting the path of the eccentric; join C C as tangents; with the centres of the eyes as centres, and the angular length to the centre of the circles as radii, form arcs cutting the circle, and the intersections are the centres of the eccentrics when loose on the shaft, from which position the correct length of the rods can be ascertained.

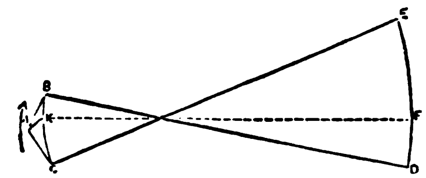
The radius of the link is the length from C to the centre of the eccentric, or the horizontal distance from the centre of the shaft to the centre of the link.

Professor Rankine, in allusion to this matter, states:—

In Fig. 240 let A be the axis of the shaft; A B, the forward eccentric radius; A C, the backward eccentric radius; B D, the forward, and C E, the backward eccentric rods; D E, the link; F, the slider or stud. Radius of curvature of link = length of rods or nearly so.

To find the motion of the slide valve produced by any intermediate position of the stud, such as F.

Fig. 240.

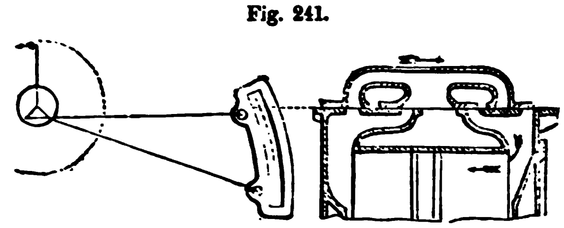


GRAY'S GEOMETRICAL DIAGRAM OF THE LINK MOTION.*

With a radius bearing the same proportion to half the distance B C, that the length of the rods B D bears to that of the link D E, draw the arc B C. If the eccentric rods are so placed (as in the figure) that when the eccentrics are inclined towards the link, the rods are crossed, make the arc B C convex towards the axis A. If the eccentric rods are so placed as not to be crossed when the eccentrics are inclined towards the link, make the arc B C concave towards A. In that arc take a point, K, dividing it in the same proportion in which the stud F divides the link D E. Then the motion of the stud, F, will be very nearly the same as if it were directly connected by a rod K F with a crank A K.

The position of the link when the eccentrics are correctly fixed on the cranked shaft affects the position of the slide valve, as illustrated by Fig. 241. In

Fig. 241.



POSITION OF LINK AND SLIDE VALVE FOR STARTING, WITH THE ECCENTRIC RODS CROSSED.

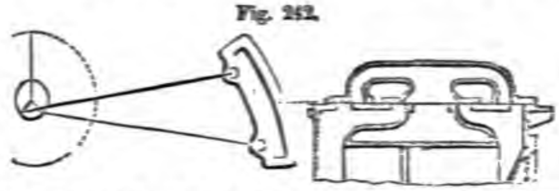
this example the link is down, the crank perpendicular, and the valve cutting off, the different arrows indicate the direction of the moving parts and the flow of the supply steam—this diagram shows the position of the valve for starting. As stopping is the next accomplishment by the link, the Fig. 242 is introduced to show the positions of the respective details. The link is now half raised; the crank in the same position as before, but the valve at half stroke or covering the ports equally. Having started and stopped the piston, the

* When the width of the small port in the slide valve exceeds the width of the opening caused by the slide, this rule must be:—Travel of valve + width of small bar + the excess alluded to.

* This construction is due to Mr. M'Farlane Gray (see his Geometry of the Slide Valve).

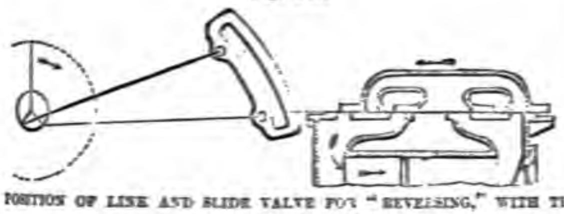
process of reversing next follows in succession, and the position of the link, crank, and valve is depicted

Fig. 242.



POSITION OF LINK AND SLIDE VALVE FOR "STOPPING," WITH THE ECCENTRIC RODS CROSSED.

Fig. 243.



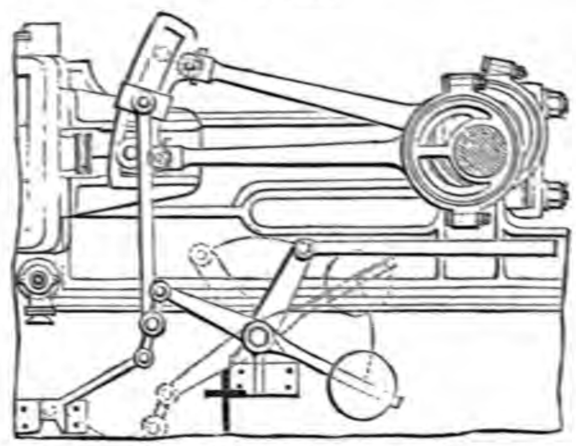
POSITION OF LINK AND SLIDE VALVE FOR "REVERSING," WITH THE ECCENTRIC RODS CROSSED.

by Fig. 243. Here it is shown that a reverse direction for the crank, piston, and valve ensues to that in Fig. 241; and also that the link is up or fully raised, while the crank is assumed to be in the same position in each instance.

Now, the position of the crank for the purposes under notice is favourable; but it may not be out of place to add that in certain positions the valve will not be affected sufficiently to start the engine, unless a slight momentum or movement is in force. With two cranks or coupled engines, in no position whatever can this disadvantage occur, simply because the cranks, being at right angles to each other, the eccentrics are similarly situated. Presume the cranks to be both at an angle of 45°, the pistons will be progressing in opposite directions; and when both links are raised or lowered, the valves will likewise affect the motion of the forward or aft engine, as the case may be.

The means adopted for suspending or supporting the link has been illustrated and described in pages 288 to 293 inclusive: the principles of the arrangement are, therefore, almost evident. The main consideration in this matter is, doubtless, the best means of acquiring a truthful motion from the rotation of the eccentric. By referring to most of the illustrations alluded to and the last three diagrams, it will be noticed that the centres of the eccentric and sliding-block are not on the horizontal line, which will be further obvious by alluding to the Fig. 239—page 354. Not only, however, is there this difference of position to consider, but also the motion described by the point of suspension. Suffice it to say, that the less the versed sine produced by the motion of the point of suspension, the more accurate will be the motion of the slide valve in relation to the action of the eccentric. Independently of the screw and sliding-block arrangement for shifting the link, Messrs. Watt have lately adopted the mode illustrated by Fig. 244—being the parallel motion originated by that firm for steam engines. The attainment in this

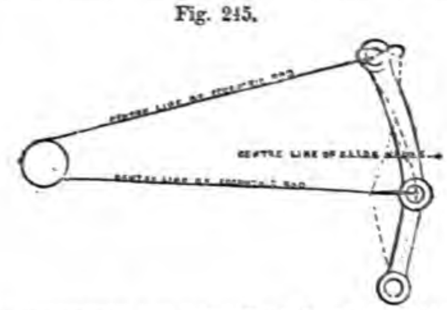
Fig. 244.



MESSRS. WATT'S LINK MOTION.

case is that the link is raised and lowered without affecting the vertical motion of the lower end of the lifting rod, or similar in effect as the screwed rod and sliding block adopted by other firms. To the present it has been stated that the stroke of the eccentric and travel of the valve should be alike, and that the point of suspension should be arranged not to affect that, if practicable. Messrs. Humphrys, however, are doubtless

Fig. 245.



RESULT OF CONNECTING THE ECCENTRIC RODS TO THE ENDS OF THE LINK BEYOND THE BLOCK PIN.*

not of this opinion, as the illustration, Fig. 245, illustrates—being a diagram of the link motion shown in

* See Burgh's Link Motion and Expansion Gear.

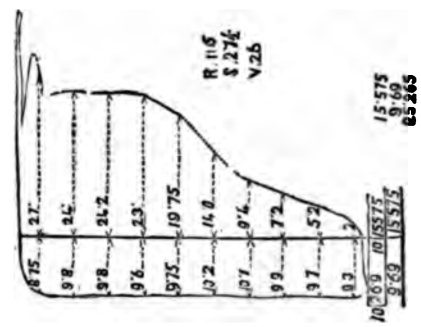
page 290. The eccentric rods being connected at the extremities of the link; the centre line of the slide valve rod between the same when the link is raised or lowered; and the point of suspension at the lower eccentric rod's connection; the travel of the valve must be less than the stroke of the eccentric. Another result also ensues; the top end of the link describes a figure, nearly approaching to an ∞; and to accomplish this, a certain amount of slip must have resulted—the form of this figure can always be determined by dividing the circle of the eccentric's path and the travel of the suspending point into an equal number of parts, and setting off relative positions on the travel of the upper end of the link.

The correct connection of the eccentric rod, when the link is raised or lowered, is at the centre of the block, by which means the least vibration or undulation results. An almost similar effect occurs by permitting the link when down to rest on the block and an outside connection; but when the link is raised a most imperfect action ensues. That the link should be suspended centrally of its length for an equal action when raised or lowered is obvious, but a sacrifice is often made, by the points of the eccentric and lifting rods being on one centre when the link is in a position for going ahead. As twin screw propulsion is now general, the link should be suspended, so that an equal action ensues when the engines are working in either direction, or the hull propelled ahead or astern.

Vertical Motion for shifting the Link.—This is the distance or chord between C C, shown in Fig. 239 —page 354—the general practice = stroke of eccentric × 2·5 to 3; using the latter numeral for short travels of the slide valve. The remaining proportions of the link, rods and eccentrics are obvious on consulting the plates and illustrations.

Indicator Diagrams.—The economical use of the steam and efficiency of the same depends greatly on the application of the slide valve and link motion, and the indicator diagram is an illustration of the result. A diagram — half-size — taken by us in the autumn of 1866 from a cylinder of a pair of marine engines is shown by Fig. 246, being introduced as an exposition. The paper being fixed on the vibrating barrel of the indicator, the steam passages freed from the condensed steam, the motion cord connected to the barrel, and the pointer lightly pressed on the paper, the atmospheric line was formed; the communication between the engine cylinder and the indicator was again opened; the motion being allowed for a moment or two without indication: all being now ready, the pointer

Fig. 246.



Half size.
MODE OF VALUING AN INDICATOR DIAGRAM.

was again pressed against the paper, and the diagram described as illustrated.

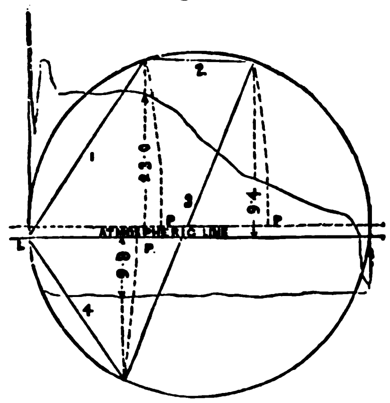
At the left hand the pointer commenced from the atmospheric line; it rose instantly to the highest point, and as quickly fell, and performed the undulation to 27·. It followed from thence in an uneven progress to 23·, when the steam was cut off. Expansion now commenced, and as a decrease in the pressure of the steam in the cylinder ensued, the pointer gradually fell from 23· to 14·, when the indented line was traced to 9·4. Expansion now ceased, and as at the same instant exhaustion commenced, the pointer gradually fell to the atmospheric line before the end of the stroke, describing below it the undulated line until the completion. On turning the corner—the return action here commenced —the pointer marked its path in an uneven line until reaching 9 8. Here compression commenced, and continued until the steam was readmitted into the cylinder termed the lead, which produced the round corner at the right hand. The pointer next ascended and joined the line it commenced. By the figure in question the action of the steam on entering and leaving the cylinder is faithfully portrayed.

Now, the veracity of a diagram is sometimes doubted; a belief perhaps circulates in the mind of the sceptic that the figure has been cooked to please the palate of the eye. The genuineness of a figure can always be known in the following manner:—Describe a circle whose diameter is equal to the length of the indicator diagram; describe at the same scale the circle of the eccentric's path; find next the angles of the eccentric, when supply, expansion, exhaustion, and compression com-

mence: from these angles the angles of the crank respectively are easily obtained, and vertical lines projected from the intersections with the main circle or path of the crank pin depict the points of supply, cut-off, exhaustion, and compression, on the diagram in question.

An example of this mode is illustrated by Fig. 247.

Fig. 247.



BURGH'S MODE OF TESTING THE TRUTH OF AN INDICATOR DIAGRAM.*

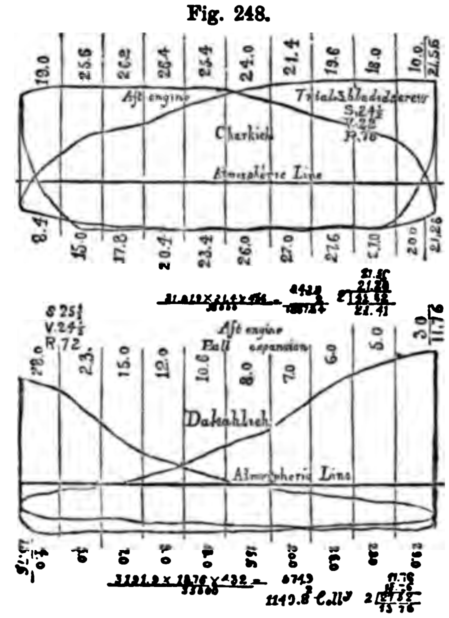
The circle denotes the path of the crank pin, and the chords 1, 2, 3, 4, relate to supply, expansion, exhaustion, and compression respectively; also the intersection with the circle depicts the positions of the crank pin, and the dotted arcs P P P show the positions of the piston, due to the length of the connecting rod. The indicator diagram is presumed to be laid on the circle, the atmospheric line intersecting with the lead point L; by which position the accuracy of the diagram, Fig. 246, is established.

It is the usual practice to indicate from both ends of the cylinder, and thus learn the action of the steam on each side of the piston. Four examples are depicted by Fig. 248, showing the cut-off by the ordinary slide valve, and the result of adopting the expansion valve, the top diagrams being the former, and those below the latter. These diagrams were taken from the engines shown in plate 22 by the Messrs. Rennie, and, as the calculations are given in detail about the figures, their relation will be obvious without description.

Another pair of diagrams is illustrated by Fig. 249, taken by Messrs. Penn from the engines of H.M.S. "Arethusa," illustrated by plate 27. It will be noticed that these examples show an almost equal cut-off on each side of the piston, a result not always produced with some engines.

It has often been argued, and with some cause too, that the adoption of the expansion valve is more whimsical than essential, and that the link motion and slide

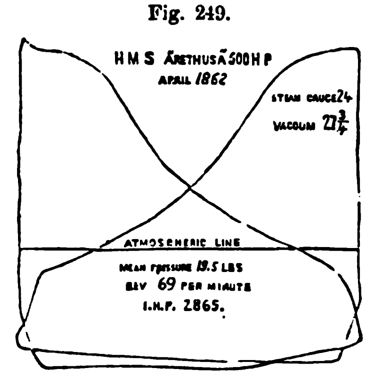
Fig. 248.



About Half size.

DIAGRAMS FROM RETURN ACTION ENGINES, BY MESSRS. RENNIE.

Fig. 249.



Half size.

INDICATOR DIAGRAM FROM DOUBLE TRUNK ENGINES, BY MESSRS. PENN.

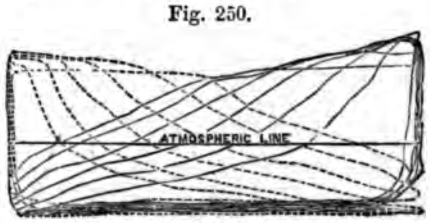
valve are sufficient for altering the points of cut-off. An illustration of a series of diagrams taken by Messrs. Napier from the engines fitted by them in the Turkish frigate "Osman Ghazy," is illustrated by Fig. 250—page 358—the respective diagrams from each end of the cylinder are depicted by plain and dotted lines, which renders obvious their relation without further notice.

Those who have taken diagrams from engines under various speeds and circumstances are aware of the results. Two diagrams, for example, taken from the same cylinder

* See Burgh's Indicator Diagram.

at different speeds of piston will often be of entirely different aspect, although the same points of cut-off may

Fig. 250.



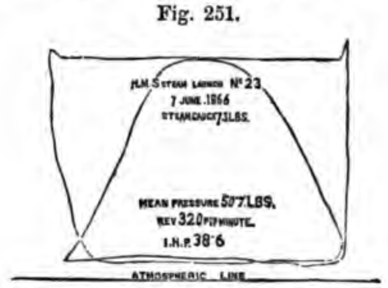
About Half size.

SERIES OF DIAGRAMS SHOWING THE EXPANSION OF STEAM BY THE SLIDE VALVE AND LINK MOTION, BY MESSRS. R. NAPIER AND SONS.

be portrayed. The cause for this variation is that the time in the slow speed diagram allowed the instrument to perform its duty without concussion, but with the fast speed diagram velocity acted against time. There are other causes also for deviations, such as the "priming" of the boilers, leakage of the joints, slackness of the string or gut, unsteadiness of the indicator, and, lastly, but not least, the clumsiness of the operator; indeed, these faults must not exist if a truthful result is required. The author has taken diagrams from the engines of a steam yacht belonging to a gallant admiral, when the piston was making 400 strokes per minute, and by properly arranging the gear a correct result was attained without a distorted figure.

Having illustrated four examples of "steam launch" engines and boilers, the following examples of diagrams are introduced. The diagrams, Fig. 251, have been taken

Fig. 251.



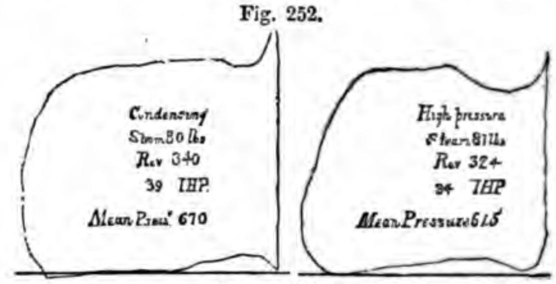
Half size.

INDICATOR DIAGRAMS TAKEN FROM "LAUNCH" ENGINES, BY MESSRS. PENN.

by Messrs. Penn; the engines and boiler being shown by plate 32. The engines and boiler in plate 29 are by Messrs. Rennie, and the diagrams—condensing and non-condensing—are illustrated by Fig. 252 from one end of the cylinder.

Diagrams taken from pumps are of equal value as those from steam cylinders, and three examples from as many pumps are represented in Fig. 253, taken by Mr. Spencer from the pumps belonging to the engines of I.S.S. "Frankfort," the arrangement and details of

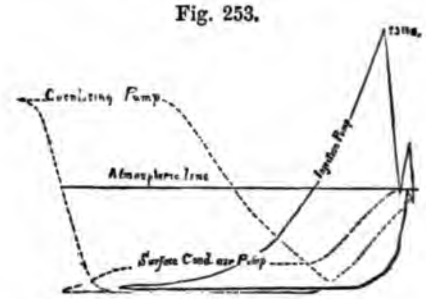
Fig. 252.



About Half size.

INDICATOR DIAGRAMS TAKEN FROM "LAUNCH" ENGINES, BY MESSRS. RENNIE.

Fig. 253.



Full size.

PUMP DIAGRAMS BY MR. SPENCER.

the condensers and engines of which are illustrated by numerous plates in this work.

The injection and circulating pump diagrams depict the pressure exerted when discharging by the form of the figure above the atmospheric line, and the traverse of the pointer below indicates the amount of exhaustion or vacuum attained, or the suction properties of the pumps.

The notes to be taken when indicating an engine are generally as follows:—

Name of ship.
Type of engines.
Nominal horse-power.
Effective area of piston in square inches.
Length of stroke of piston in feet.
Name and locality of trial.
Time and date of ditto.
End of cylinder (top or bottom, back or front).
Number of strokes of piston per minute.
Grade of expansion in gear.
Pressure of steam indicated by the gauges in the engine and boiler rooms.
Exhaustion indicated by the vacuum gauge.
Amount of injection water passage opened.

Feed water, off or on.
Temperature of the engine and stoking-rooms.
State of fires in the boilers.
Height of water in ditto.
Priming, if any symptoms.
State of the working, bearings of the engine, and shafting from the forward main bearing to the stern tube stuffing box.
General appearance of the duty of the engine.
Type of screw-propeller and particulars.
Speed of the ship in knots per hour.
State of the water and tide.

Indicated Horse-power.—Let E = effective area of piston in square inches; M = mean pressure of steam and vacuum attained, as indicated; S = speed of piston in feet per minute; then

$$\frac{E \times M \times S}{33{,}000} = \text{I.H.P.}$$

Strains—Tensile and Compression.—The connecting rod of an engine is subject to certain strains—direct and angular being the most prominent, and these act with contrary effects, *i.e.*, compression and tensile. The direction of the circuit of the crank pin also affects the lines of the strains, inasmuch that if the direction is reversed, so are the strains. To exemplify—presume the diagram, Fig. 254, to illustrate

Fig. 254.

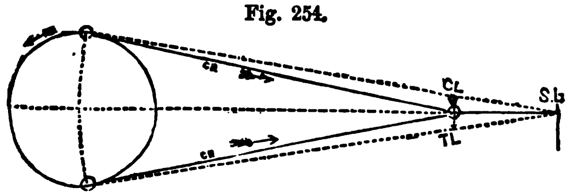


DIAGRAM OF THE STRAINS OF THE CONNECTING ROD.

the circle of the path of the crank pin, angles of the connecting rod at the half stroke of the piston, and the position of the stuffing box of the piston rod. Now, lines drawn from the stuffing box S B to the centre of the crank pin indicate the lines of strain; C L being above the centre line of motion, and T L below. When the connecting rod C R is forcing the crank pin, the effect is to disturb the inertia, but the virtual resistance may be said to be in a reverse direction, as depicted by the arrow above the centre line; the connecting and piston rods being attached at C L—compression line—the strain is in the direction of the downward arrow, or to descend from C L. When the crank returns on the lower half of the circle, the steam acting also reversely, the strain is reversed; here also is no actual resistance, but a decided pull or tensile strain occurs as shown by T L. The point of the rod's connection inclines towards the line of strain T L, and thus the friction on the face of the guide block is the same at each half stroke of the piston. The illustration, Fig. 255, is introduced to

Fig. 255.

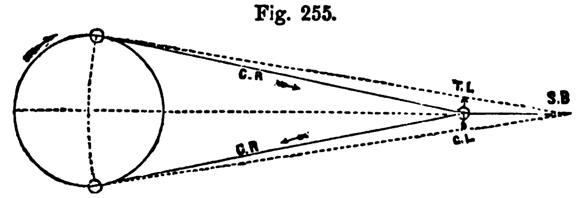


DIAGRAM OF THE STRAINS OF THE CONNECTING ROD.

further assist conception—being a diagram showing the result of reversing the direction of the crank pin's motion. In this case T L is above the centre line of motion, and C L below, and the arrow between C L and T L ascending from and to the lines of strain, the arrows behind C R, C R indicate the reverse direction of the connecting rod above and below the line of motion, and S B is the stuffing box as before.

Not only is the connecting rod exposed to the strains alluded to, but also the pump, slide valve, eccentric, piston, and other rods, receiving an alternate motion. The details partaking of tensile strain only are the bolts of the connecting rod, eccentric, trunk gudgeon, guide block, and main frame cap, also the studs securing the cylinder cover, boring hole cover, and pump doors.

Strains—Shearing and Torsion.—The crank shaft, screw shaft, starting wheel shaft, lever weigh-shaft, and all rods or shafts that vibrate or rotate, are subject to a torsion strain. The shearing properties bear a relation to the crank pin, coupling bolts of the screw shaft, trunk gudgeon, guide block crosshead, link pins, sliding block pin, and all details, receiving a lateral strain at right angles to the longitudinal line of repose or support. It may be added that the crank shaft partakes of the shearing strain also when the cranks are on the horizontal line, or in a line with the centre line of motion. The strain imposed and to be resisted can always be known by remembering the requirement and locality of the detail.

Frictional Surfaces.—The moving friction of an engine has been estimated in some instances as 10 per cent. of the actual horse-power, but the only means of

knowing the actual resistance or power requisite to overcome the inertia is to calculate the area of the frictional surfaces and define the nature and effect of the strains exerted on them. Now the nature of friction is due to the direction of the moving surface, in relation to the supporting surface; also the grain or fibre of the metals should not be neglected to attain a correct result. It is obvious that with the marine engine, the motions most frequent are sliding and rotating,—vibrating being only in relation to two details. The portions partaking of a sliding motion are the steam and pump pistons, trunk and piston rods, stuffing boxes and glands, piston rod guide block, channel surface, slide valve and expansion valve rods, stuffing box and gland, slide valve rod guide, stuffing box and gland of feed and bilge pump, plungers, gravity of water, and passage through valves.

The rotating details are the crank shaft bearings, crank pins, eccentrics, expansion gear, thrust block, plummer block, stuffing box, stern tubing, and propeller bearings—the friction of the latter is increased when the propeller overhangs. The vibrating motion is confined to the link motion and suspending pins.

Exceptional to the quantity of the vibrating surface, the preceding remarks apply to oscillating engines also; but as with this type the cylinders and lever blocks vibrate, the result of that movement exceeds that for screw engines.

Table of co-efficients of the friction, with unguents of the ordinary kind, of sliding surfaces :—

Surfaces in Contact.	Co-efficients.		Unguents.
	Starting.	Motion.	
Cast iron upon cast iron . .	0·1000	0·100	Tallow.
Ditto	0·064	0·064	Olive oil.
Cast iron upon brass	..	0·103	Tallow.
Ditto	..	0·078	Olive oil.
Brass upon cast iron	0·106	0·086	Tallow.
Ditto	..	0·077	Olive oil.
Brass upon wrought iron . .	..	0·081	Tallow.
Ditto	..	0·072	Olive oil.
Brass upon brass	..	0·058	Olive oil.
Wrought iron upon brass . .	..	0·078	Olive oil.
Steel upon brass	..	0·056	Tallow.
Ditto	..	0·053	Olive oil.
Wood upon iron	..	0·085	Tallow.
Wood upon wood	0·254	0·083	Tallow.

With reference to the friction of shafts in motion, M. Morin has deduced as follows :—

FRICTION OF SHAFTS IN MOTION.

Designation of surface in contact.	Dry or slightly greasy, or wet.	Oil, Tallow, or Hog's Lard.	
		Supplied in the ordinary manner.	The grease continually running.
Brass on brass	..	0·079	..
Brass on cast iron	..	0·072	0·049
Iron on brass	0·251	0·075	0·054
Iron on cast iron	..	0·075	0·054
Cast iron on cast iron	0·137	0·075	0·054
Cast iron on brass	0·194	0·075	0·054
Iron on lignum vitæ	0·188	0·125	..
Cast iron on lignum vitæ . .	0·185	0·100	0·092
Lignum vitæ on cast iron . .	..	0·116	0·170

Professor Rankine, in allusion to this subject, states: "To calculate the moment of friction of an axle, multiply the resultant load by the radius of the axle, and by the sine of the angle of repose."

This subject has been treated briefly also by Mr. Nystrom in his Pocket-Book, thus :—

Letters denote.

a = Fibres of the woods that are parallel to themselves, and to the direction of motion.

b = Fibres at right angles to fibres.

c = Fibres vertical on the fibres which are parallel to the motion.

d = Fibres parallel to themselves, but at right angles to the motion, length by length.

e = Fibres vertical, end to end.

Example. A vessel of 800 tons is to be hauled up an inclined plane, which inclines 9° 40′ from the horizon; the plane is of oak, and greased with tallow. What power is required to haul her up?

The co-efficient for oak on oak with continued motion is $f = 0{\cdot}097$, say $0{\cdot}1$, then,

$800 \times \sin. 9° 40' = 800 \times 0{\cdot}16791 = 124{\cdot}328$ tons,

the force required if there were no friction, and

$800 \times \cos. 9° 40' \times 0{\cdot}1 = 800 \times 0{\cdot}9858 \times 0{\cdot}1 = 78{\cdot}864$ tons,

the force required for the friction only, and

$$\begin{array}{r} 124{\cdot}328 \\ 78{\cdot}864 \\ \hline 203{\cdot}192 \end{array}$$

203·192 tons, the force required to haul her up.

The effect lost by friction in axle and bearing is expressed simply by the formula

$$P = \frac{\pi d W n f}{12 \cdot 60} = \frac{W d n f}{230},$$

in which W = the weight of pressure in the bearing, d = diameter on which the friction acts in inches, n = number of revolutions per minute, and f = co-efficient of friction. In common machinery kept in good order, the co-efficient of friction can be assumed to $f = 0 \cdot 065$, then

$$P = \frac{W d n}{3530}, \qquad H = \frac{W d n}{1941500}$$

Example. The pressure on a steam piston is 20000 pounds, and makes $n = 40$ double strokes per minute. Required the friction in the shaft of $d = 8$ inches.

$$H = \frac{20000 \times 8 \times 40}{1941500} = 3 \cdot 3 \text{ horses, the loss by friction.}$$

Friction in Guides.

W = pressure on the steam piston in pounds.
S = stroke of piston in feet.
l = length of connecting rod in feet.
H = horse-power of the friction.

$$H = \frac{W S n}{350000 \sqrt{5 l^2 - S^2}}$$

Example. The pressure on a steam piston being W = 30,000 pounds, stroke $S = 4$ feet, length of connecting rod $l = 7$ feet, and making 50 revolutions per minute. Required the horse-power of the friction H.

$$H = \frac{30000 \times 4 \times 50}{350000 \sqrt{5 \times 7^2 = 4^2}} = 1 \cdot 13 \text{ horses.}$$

R. Mallet, Esq., C.E., when alluding to the result of adopting a *short* connecting rod, states: "The *mechanical* disadvantages of the shorter connecting rod are presumed to be comprehended under—1st, Increased friction, produced by the increased rubbing pressure of the guide piece, that secures the movement parallel to itself of the piston rod head; 2nd, A certain amount of increased friction at the joint pin between the piston and connecting rods; and 3rd, A certain amount of increased friction at the crank shaft bearings, due to the alternate lift and pressure down upon the shaft."

The two last sources of increased friction may be passed over as *very* small.

The *increase* of friction due to the first, at its maximum, viz., when the connecting rod makes its largest angle with the path of the piston rod, will be very nearly in the ratio of the length of the connecting rod, divided by that of the crank.

Let us consider what this would amount to in an engine delivering 300 horse-power.

The area of cylinder being taken at 4536 square inches, with 20 lbs. mean pressure, gives a total of 90720 lbs. pressure. If now the connecting rod be six cranks length, $\frac{90720}{6} = 15120$ lbs. will be the greatest pressure upon the guide; and as the co-efficient of friction of brass on iron when lubricated is $0 \cdot 0909$ the pressure; then the increased resistance to the piston's motion, due to this cause, over and above that of an engine with an *infinite length* of connecting rod, will be $\frac{15120}{11} = 1374$ lbs. In the same way, with a connecting rod of only three and a half times the length of the crank, $\frac{90720}{3 \cdot 5 \times .0909} = 2356$ lbs. is the corresponding increased resistance to the piston. The difference, or $2356 - 1374 = 982$ lbs., is the disadvantage incurred by the shorter rod; but this is the maximum disadvantage, which is zero, when the crank is on either centre; so that the mean resistance is only $\frac{982}{2} = 491$ lbs., over and above that of the connecting rod of even *six cranks* in length, which is longer than employed in any horizontal marine engine; and if this be diffused upon the above area of piston, it will be about 1-9th of a pound per square inch, or 1-180th of the total pressure per inch of piston. This quantity is extremely small—so small that, in the same example, if nine inches diameter were sufficient for the connecting rod, if always in line with the piston—*i. e.*, infinite in length, it would require, when reduced to three and a half cranks in length, to be made about $9 \cdot 1874$ inches diameter for equal pressure on the unit of section. This very simple consideration appears to render it extremely doubtful whether, in the case of the trunk type of engine, the increased length of connecting rod be not achieved at the expense of an appreciable *loss*, in place of any gain in power; from the friction of the trunk, gripped by its packing, over a very extensive surface, the degree of pressure upon which to produce stanchness, can only be at any time conjectured. And this friction is, in relation to the length of the rod, increased by the oblique pressure, just as is that of the piston guide in the former case; so that in our judgment

simplicity and come-at-ability of all the working parts should be held paramount to the question of whether the connecting rod be three and a half, or be four or five cranks in length.

Strength of Materials.—The table devoted to this important subject is a *bonâ fide* compilation from the mean of the results of experiments by Rennie, Fairbairn, Napier, Maudslay, Mallet, and Telford.

Name.	Breaking *Tensile* Strain in lbs. per square inch.	Disturbing *Compression* Strain in lbs. per square inch.	Dividing *Shearing* Strain in lbs. per sq. inch.	Twisting *Torsion* Strain in lbs. per sq. inch.	Ditto for bar, one inch diameter.
Wrought iron.	56,000	36,000	50,000	15,360	12,063
Steel	100,000	..	..	25,497	20,025
Cast iron. . .	20,000	110,000	27,700	15,206	11,943
Gun metal . .	35,000	10,350	..	8,000	6,000
British oak. .	10,000	10,000	2,300	2,350	..
Red pine . . .	13,000	6,000	600	1,540	..

The author personally received from G. B. Rennie, Esq., C.E., the following particulars of experiments by the late G. Rennie, Esq., C.E., and also some note of American Practice on the resistance of Metals to Torsion.

"Mr. George Rennie, found by experiment, 1 inch square bar bore 112 lbs. at 3 feet radius, or 336 lbs. at one foot; wrought iron bar is taken 14 : 9; then, 1 inch square bar wrought iron at one foot radius = 984 lbs. American experiments—good castings—1 inch diameter at 1 foot radius will break at 583 lbs.

"Wrought iron 1 inch diameter at 1 foot radius distorts without breaking, 642 lbs., but begins to yield at —permanent load—300 lbs."

Application of Materials.—Before a correct formula can be deduced the *permanent load* must be known, *i.e.*, the weight of the moving material, and the nature of the motion. Now, all rods subject to compression and tensile strains are connected at the extremities; and the load on them is the weight, or inertia of the connecting portions. For example, the piston rods are subject to the strains alluded to, and the load consists of the piston at one end and the crosshead at the other; obviously the length between these details affects the sectional area of the rod. The connecting rod is hung between a sliding and rotating motion; consequently, the load is in some measure due to the length of the rod in proportion to the radius of the circle described. In the first case, the sliding point has a load on it due to the weight of the piston rods beyond the stuffing boxes, crosshead, guide block, bolts and nuts, and a portion of the weight of the connecting rod—the latter mostly when it is ascending from the full stroke; in the second instance, the rotating surface is affected by the weight of the rod, the weight of the cranks, or centrifugal force and inertia of the shafting throughout the bearings. It is for these reasons that makers differ in their proportion; and when an engineer correctly defines the least proportions, he is justified in adopting them due to the reduction of the permanent load. Reverse to this, it is possible to design an engine with such maximum proportions that the inertia exceeds the pressure of the steam on the piston, or similar to constructing a girder that cannot bear its weight, or the permanent load exceeding the resistance to fracture.

It is obvious from these conclusions, that when the sectional area of the piston rod is required, the pressure of the steam must be noticed first; next, the effective area of the cylinder; thirdly, the load and the length of its travel; fourthly, the compression strain on the rod; and, lastly, the factor of safety. The sectional area of the connecting rod is exposed to nearly the same strain as the former detail, but the load is obviated, and the inertia of the crank shafting and propeller takes its place.

Professor Rankine treats inertia, when requiring to know the power to disturb it, as follows:—

To reduce the inertia or mass of the machine to the driving point. Multiply the weight of each moving portion of the machine by the square of the ratio of its velocity to the velocity of the driving point; and add together the products; the sum will be the weight of the mass which, if concentrated at the driving point, would require the same force to produce a given change in its speed, in the course of a given time or of a given motion, that is required by the actual machine.

The crank shaft's diameter is determinable from the length of the crank as a lever, and the total pressure on the piston as a weight on the end of the same, the torsion strain being the divisor. Now, as the strength of shafts are in proportion to the cubes of their diameters, the sectional area, therefore, is not often alluded to in the formula.

In allusion to oscillating paddle engines, the proportions are widely different to screw engines, simply because the permanent load and its travel settles the pro-

portion of the piston rod, as much as the pressure of the steam.

Sectional Area of Piston Rod = *effective area of cylinder in square inches* × *pressure of steam in pounds per square inch* × [*permanent load in pounds* × *stroke of the piston in feet*] ÷ *compression strain in pounds* × *factor of safety*—(*area for two rods* = *preceding area*).

Sectional Area of Connecting Rod.—The correct mode of deducing this section is to consider the angle of the rod, inertia of the shafting and propeller, and number of revolutions per minute. In practice, however, the diameter of the piston rod and the connecting rod are often the same, and universally so when the length of the latter between the centres of connection = *throw of crank* × 3·5. Should this length be exceeded, which it is with return-action engines, the maximum area of the rod in question = *area of the piston rod* + 3 *square inches per foot of the length of the turned portion of the connecting rod.* The ends are often reduced to the diameter of the piston rod; the increased area being preserved only at the centre of the length. It may also be added that symmetry has much to do with this detail—the piston rod being the main determination.

Sectional Area of Connecting Pin.—This detail is supported by the guide block, and secured in each branch or fork of the connecting rod. The strain from the piston rod is transmitted to the centres of the forks, and from thence to the centre of the connecting rod. The pin, therefore, receives shearing and deflecting strains — the former being at the ends of the bearing, and the latter beyond the same. Now, the certainty that the deflection of the pin must disturb the securing bolts of the guide block before the rupture occurs, prompts the idea that the shearing strain is the main consideration in this example when the length of the bearing of the pin equals its diameter. The formula is, therefore, as follows:—*Effective area of the piston in square inches* × *pressure of the steam in pounds per square inch* + [*permanent load in pounds* × *stroke of the piston in feet*] ÷ *shearing strain in pounds* × *factor of safety.* This produces a diameter less than the single piston rod; the practice of many engineers, however, is to make the diameter of both these details alike, and the length of the bearing of the pin in the block = *diameter* × 1·5.

Sectional Area of Crosshead for double Piston Rods.—In this case the detail in question can be treated as a beam supported at the extremities, and the load centrally situated for a certain length. The piston rods form the supports, and the force from the piston rods transmitted to the centre, and met by the resistance, is the load. Now, the strains imposed are shearing at the ends of the bearing, deflection beyond, and compression and tensile combined when a rupture occurs.

It is well known that when a bar of wrought iron is broken across a certain portion is torn asunder, while the remainder is disturbed only, or compressed, without a perfect division. Now, the crosshead is partially exposed to the same result of fracture, *i.e.*, if the detail does not bend, it will be sheared, and if not sheared, the side receiving the force will be compressed, and that opposite extended, or torn asunder; from which result it is evident that the shearing strain must be resisted and the distance between the supports observed to prevent deflection. The correct formula, then, is resolved thus:—*Effective area of the steam piston in square inches* × *pressure of the steam in pounds per square inch* + [*permanent load in pounds* × *stroke of the piston in feet*] ÷ *shearing strains in pounds* × *length in feet of crosshead between centres of piston rods* × *factor of safety.*

The permanent load is considered the same throughout these formulæ, as the weight in motion is unalterable. The factor of safety also is considered alike in each instance. Now, the sums of these two functions are dependent on the weight of the material and velocity of the piston. With the load the travel must be noticed, due to the fact, that the amount of matter shifted or lifted a certain distance in a given time requires a proportionate power, and the material employed a relative resistance to prevent fracture. Obviously, then, if the load moves through one foot in a second, the power and resistance are proportionate to the speed; and if the same weight moves through 3 feet in a second, the power and resistance are also increased. It is, therefore, for this reason that the *load* × *travel* is introduced in the formula. Next, as to the factor of safety, — this multiplier determines the increase of sectional area proportionate to the maximum speed and strains. It also bears a relation to the load and its travel, inasmuch that if the speed is increased, the load will affect the area of the rods in question. The weight of the material comprising the load differs considerably in each type of engine by various makers. The follow-

ing table of the load and factor of safety is the result of the mean of several examples :—

Type of Engine	Nominal H.P. collectively.	Load = lbs. per sq. inch × area of cylinder.	Maximum speed of piston in feet per minute.	Factor.
Double Trunk	200 to 1500	4·5 to 2·5	1,000 to 1,800	6 to 10
Direct acting	200 to 1500	6 to 4	1,000 to 1,800	9 to 14
Return acting	200 to 1500	8 to 5	1,000 to 1,800	10 to 15

The sectional areas of the pump, slide valve, and eccentric rods, are deduced from the same formula as the piston rods, remembering the permanent load in connection. The link pins and all portions having a lateral stress must be considered as the crosshead, being virtually beams with a central load. The securing or cap-bolts of the connecting rod and main frame are subject to a tensile strain; but when the cap-bolts of the main frame are at right angles with the central line of motion, a shearing strain is imposed. It is obvious, therefore, that any detail of an engine can be correctly proportioned without doubt, by considering the direction of the strains, the permanent load, and the material employed.

Crank Pin, Cranks, and Shafts.—The area of the crank pin = *effective area of the steam piston in square inches* × *pressure of the steam in pounds per square inch* + [*permanent load* × *length of piston's stroke in feet*] ÷ *shearing strain in pounds* × *factor of safety* × *proportion of bearing's length to the virtual diameter*. The length of the bearing or distance between the cranks = *virtual diameter* × 2. For screw engines the diameter of the crank pin and shaft are the same, due to construction rather than correct proportion.

The sectional area of the crank depends on the form and length between the centres of the pin and shaft. With screw engines, construction is observed rather than proportion; the rule generally is, when the *width of the crank equals the diameter of the shaft, the thickness is about three-fourths of the diameter*. The *correct* proportion is known by considering the cranks as beams supported at one end, and the load at the other extremity; the factor of safety depending on the form of the section. The length = throw of the crank; the load = pressure of the steam on the piston in pounds + permanent load and its momentum effect, and the sectional area should be proportioned to the length. With paddle engines the area of each crank is much less than for screw engines, on account of the cranks being forged separately and keyed on the shaft. The average proportion is; *width of the web = diameter of crank shaft*, and the *thickness = half the diameter;* the width being tapered from the main boss.

The shaft of an engine is the transmitter of the power developed and expended at its extremities. The force exerted by the steam is thrown on the crank pin, from thence to the crank, and follows on through the shaft to the opposite end, where the propeller receives it. Now, evidently the strains imposed are not alike; the pin is subject to a shearing strain, the crank deflecting or bending, and the shaft to torsion and shearing strains; and when the latter is exerted, the crank is subject to a compression and tensile strain. The portion between the outer crank and the propeller is exposed to an elongated twisting strain, rather than a direct torsion. With the screw, compression and tensile strains are in force beyond the thrust block; and, therefore, to deduce a correct formula, the type of propeller must be noticed. Another cause for the last consideration is that the area of the propelling agent when acted on by the sea, tends to hold the engine; and thus two forces are straining the shaft in opposite directions simultaneously.

In most cases it is preferred to represent the length of the shaft, and all contingent circumstances, by a constant number or factor of safety. The practice of T. B. Winter, Esq., C.E., has kindly been put at the author's disposal, being as follows. Let

x = *diameter of crank shaft.*
A = *total pressure in pounds on the piston.*
L = *length of crank in inches.*
C = ·000004 *as constant number.*
F = *factor of safety.*

Then—

$$x = F \times \sqrt[3]{\cdot 000004 \times A \times L}.$$

Other engineers adopt the following; when
D = *diameter of shaft in inches.*
P = *total pressure on piston in pounds.*
L = *length of crank in inches.*
T = *torsion strain in pounds for round shafts.*
F = *factor of safety.*

Then—

$$D \sqrt[3]{\frac{P \times L}{T} \times F}$$

Now, when considering the sum of the total pressure, it must be remembered that both pistons, at cer-

tain positions, are affecting the crank pins simultaneously in opposite directions; and, therefore, it is not erroneous to recognise the united areas of the pistons. When the propeller overhangs the bearing of the shaft, the area of the latter should be enlarged, as the weight of the propeller acts as a load in constant motion.

The screw shaft should be supported in proportion to its length and diameter, or the distance between the "plummer blocks" should not exceed 12 diameters; the general practice being 8 to 10 diameters. By supporting the shaft correctly, and the adoption of two thrust-blocks—one at each end of the "screw alley," the contingent and constant strains on the shaft are more effectually resisted.

Condensers and Pumps.*—The proportions of these portions are obvious from the various examples quoted in this work. The principles to be noticed are the pressure of the steam in the cylinder, and the lowest temperature available; the cubical contents of the steam employed at each stroke of the piston, which amount is actually what must be condensed at the same time; the temperature of the circulating water, and the amount requisite for each volume of steam, remembering that the *water absorbs* the *caloric.*

The general practice is to consider the superficial area for the tubes and cubical contents for the pumps in relation to the cubical contents of the cylinder. It will be remembered that the circulating water is sometimes forced through or amongst the tubes by the centrifugal pump, separate motive power being often adopted; the theory of the centrifugal pump has been treated at some length, by Joseph Glynn, Esq., F.R.S., in his work on the "Power of Water." He states:—

Of the total work employed in producing rotation, that portion which represents the force in the normal or the centrifugal force is that alone which under any circumstances can become duty in the centrifugal pump.

Generally—

Let G = the weight of a revolving body; and hence its mass $M = \frac{G}{g}$, g having the usual relation to gravity.

r = the radius of revolution.

v = the velocity of revolution in the circumference.

P = the force in the normal, or the centrifugal force.

* See Burgh's "Condensation of Steam."

$$\text{Then } P = \frac{M v^2}{r} = \frac{G v^2}{g r} = 2\frac{v^2 G}{2 g r}\text{; hence}$$

$$P : G :: 2\frac{v^2}{2g} : r.$$

That is, the centrifugal force is to the weight of the body in revolution as twice the height due to its velocity is to the radius of revolution.

The resistance being uniform, we may express v in terms of the time of revolution T with the radius r, and

$$P = \left(\frac{2\pi r}{T}\right)^2 \frac{M}{r} = \frac{4\pi^2}{T^2} M r = \frac{4}{g T^2} G r;$$

and as the constant $4\pi^2 = 39 \cdot 4784$,

$$P = \frac{39 \cdot 4784}{T^2} M r = 1 \cdot 224 \times \frac{G r}{T^2};$$

or if n = the number of revolutions per minute, so that $T = \frac{60''}{n}$, then

$$P = \frac{39 \cdot 4784}{3600} \times n^2 \times M r = \cdot 010966\, n^2 \times M r,$$

$$\text{or } P = \cdot 000331 \times n^2 \times G r.$$

Lastly, as $\frac{2\pi}{T} = \omega$, the *angular* velocity,

$$P = \omega_2 \times M r = \omega^2 \times \frac{G}{g} r.$$

These various expressions for P become convenient for all calculations in which centrifugal force enters.

Where the revolving body is a fluid, and a particle whose weight is G is transferred by the normal force from the axis of rotation to the extremity of the radius r, then the work done, L, is—

$$L = \frac{\omega^2 r^2}{2g} \times G = \frac{v^2}{2g} G,$$

v being the velocity of rotation at the *extremity* of the radius r. Or if the particle start not from the axis, but from some intermediate point in a radius, then

$$L = \left(\frac{v_2 - v_{12}}{2g}\right) G.$$

In the case of a properly proportioned centrifugal pump—

Let $C R = r_1$.

v_1 = the surface velocity of the vanes, which must be proportioned to

H = maximum dynamic head of water to be overcome, and which consists of

z = the elevation to which the water is to be delivered from the lower level.

h = the height due to the velocity of delivery.

h_1 = the head lost in overcoming resistances in the machine.

$$\text{Then} \frac{v_1{}^2}{g} = H = z + \frac{V^2}{2g}(1 + \Sigma f),$$

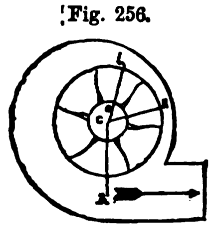
V being the velocity in the ascending main of the pump, and Σf the sum of the several resistances; and the surface velocity of the blades is—

$$v_1 = \sqrt{\left(g z + \frac{V^2}{2}\left(1 + \cdot 025 \frac{d}{z}\right)\right)}$$

d being the diameter of the pipe when it is wholly vertical, and therefore its length $l = z$; but when otherwise for $\cdot 025 \frac{z}{d}$ we must substitute $\cdot 025 \frac{l}{d}$.

Let d, = the diameter of the ascending main, be taken as unity. Then in proportioning the pump, let the external radius of the blades, C R (fig. 256), = $\frac{1}{4} d$;

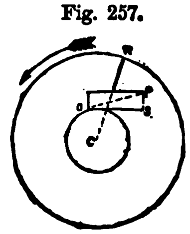
Fig. 256.



the radius of the ears of the pump = $\frac{3}{4} d$; and the diameter of each of the indraught passages = d. The breadth of the blades = $\frac{1}{4} d$ nearly, and the mean radius of the casing of the pump = $\frac{1}{4} d \times \frac{v_1}{V}$ = C A, fig. 256.

The fan-blades should be perfectly radial at the outer extremity, and for at least one-half their length. The inner portion should be curved, as in fig. 257, forwards

Fig. 257.



in the direction of revolution of the fan, and should so reach the inner edge of the revolving disc of blades, that the angle $p\ o\ s$, = β, should be—

$$\beta = \frac{V_0}{V_2}$$

V_0 being the radial velocity of the water, and V_2 being that of the inner edge of the fan-blade.

As regards the power required to drive a centrifugal pump, and to raise per minute a given weight of water, W, it may be taken at

$$2\,W\left(z + \frac{V^2}{2g}\left(1 + \cdot 025 \frac{l}{d}\right)\right)$$

for very few of such pumps in reality return in duty more than fifty per cent., and the great majority far less.

Propellers — Paddle and Screw.— Commencing with the paddle-wheel, attention must be directed to the illustration in page 240. The mode of setting out the floats, or the pitch, is as follows:—Determine the number of floats to be immersed at the same time; which conclusion is due to the area of each float, in proportion to the friction of the vessel to be propelled. This can be more readily appreciated by remembering that the weight of the vessel and its form defines the friction to be overcome; and also that momentum assists the vessel in her progress when the inertia is fully overcome, or what is often termed the way in action. It is obvious, then that the areas of the floats immersed simultaneously are the propelling agents of the vessel, and therefore the starting power must be the main consideration. For tug steamers a broad float is preferred, because speed is sacrificed to power; but with steamers with fine lines, narrow floats are adopted, because the momentum shall not be retarded by the width of the propelling surface. Another mode of settling this question is, to recognize the maximum area of the displacement, proportionate to the length of the immersed portion in front of the paddle-wheel, by which the resistance to be overcome is known. But while acknowledging these facts, it must be remembered that, if the section of the displacement is enlarged behind the paddle-wheel, a drag results, and thus the momentum is impeded.

Professor Rankine's views on the subjects of displacements of hulls, friction, and paddle and screw propulsion, are expressed in his "Rules and Tables," thus:—

"Given the intended greatest speed of a ship in knots;

to find the least length of the *after body* necessary, in order that the resistance may not increase faster than the square of the speed; take *three-eighths* of the square of the speed in knots for the length in feet.

To fulfil the same condition, the *fore body* should not be shorter than the length for the after body given by the preceding rule, and may with advantage be $1\frac{1}{2}$ times as long.

To find the greatest speed in knots suited to a given length of after body in feet; take the square root of $2\frac{2}{3}$ times that length.

When the speed does not exceed the limit given by the above, to find the probable resistance in lbs.; measure the *mean immersed girth* of the ship on her body plan; multiply it by her length on the water line; then multiply by 1 + 4—mean square of sines of angles of obliquity of stream lines.—The product is called the *augmented surface.* Then multiply the augmented surface in square feet by the square of the speed in knots, and by a constant co-efficient; the product will be the probable resistance in lbs.

Additional Resistance of Ship, due to short after body.—Let v be the speed in knots; l, the proper least length of after body, in feet $= \frac{3}{8}v^2$; l', the actual length of after body; S, the area of midship section, in square feet: $\sin^2\gamma$, the mean of the squares of the sines of the angles of obliquity of the stream lines of the after body; then, additional resistance in lbs.—

$$= 5 \cdot 66 \sin 2\gamma \cdot S \sqrt{\left(1 - \frac{l'^2}{l^2}\right)}, \text{ nearly.}$$

Co-efficient for clean painted iron vessels, ·01:

„ for clean coppered vessels ·009 to ·008;

„ for moderately rough iron vessels, ·011 and upwards.

For an approximate value of the resistance in well-designed steamers, with clean painted bottoms; multiply the square of the speed in knots by the square of the cube root of the displacement in tons. For different types of steamers, the resistance ranges from ·8 to 1·5 of that given by the preceding calculation.

To estimate the *net* or *effective horse-power* expended in propelling the vessel; multiply the resistance by the speed in knots, and divide the product by 326.

To estimate the *gross* or *indicated horse-power* required; divide the same product by 326, and by the combined *efficiency* of engine and propeller. In ordinary cases that efficiency is from ·6 to ·625—average, say ·613: therefore in such cases the preceding product is to be divided by 200.

Thrust of Propellers.—To calculate the thrust of a propelling instrument—jet, paddle, or screw—in lbs.; multiply together the transverse sectional area, in square feet, of the stream driven astern by the propeller; the speed of that stream, *relatively to the ship*, in knots; the *real slip*, or part of that speed which is impressed on that stream by the propeller, also in knots; and the constant 5·66 for sea-water, or 5·5 for fresh water.

Given, the product of the velocity of advance, in knots, of a screw propeller as if through a solid (= pitch in knots × revolutions per hour) into the slip of that screw relatively to the water in which it works (also in knots); required the product of speed and slip of the stream from the screw, for use in the above Rule. Multiply the first product by $1 - \frac{\cdot 8 \text{ pitch of screw}}{\text{circumference}}$. This is a good rough approximation when the circumference is between $1\frac{1}{2}$ and $3\frac{1}{2}$ times the pitch.

The speed of the stream driven astern by feathering paddles is sensibly equal to that of their centres; by radial paddles, to that of their outer edges. The gross power required to drive a radial paddle-wheel is greater than that required to drive a feathering paddle-wheel of equal thrust, in the ratio of

$$\sqrt{\left(\frac{\text{outer radius of wheel}}{\text{height of axis above water}}\right)}, \text{ nearly.}$$

Geometry of the Feathering Paddle Wheel.—An illustration of the relative positions of the floats is depicted by Fig. 70—page 240. To define the throw of the eccentric is the first consideration, which, in general practice, is thus:—

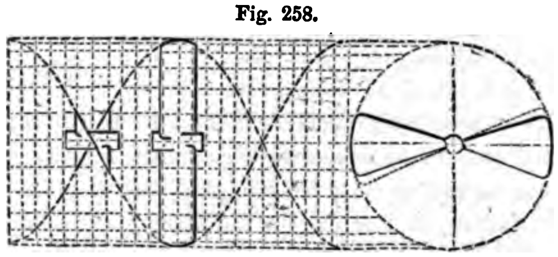
Produce a perpendicular line through the centre of the paddle shaft; describe thereon the centre of the "polygon, or the centre line of the lever shafts; determine the position and length of the lever of the lowest float; connect the top intersection of the perpendicular line and the polygon, by an angular line passing through the centre of the small eye of the lever in question; next draw a line at right angles with the angular line passing through the centre of the shaft; and the intersection with the angular line is the centre of the eccentric.

The position of each float can be readily ascertained by describing a circle equal to the diameter of the polygon

on the centre of the eccentric; and as the pitch of the floats and lever shafts is equal, and the length of the levers alike, the points of intersection on the eccentric circle are obvious; also the angles of the floats. The pitch of the centres of the lever rods on the eccentric is equal in this case; but in some instances the pitch is unequal, being proportioned from the inequality of the pitches of the lever's centres on the eccentric circle.

Geometrical Delineation of the "Common Screw Propeller."—The screw is simply a raised surface around the circumference of a cylinder, the ends of the surrounding portion being unconnected, due to the spiral form or longitudinal advance of the curve. Now the screw propeller, in its ordinary application, is the same in principle; or the pitch of the thread, its depth and length, determine the form of the blades. As an evidence of this law, the illustration, Fig. 258, is introduced, being an example in actual practice: the pitch is

Fig. 258.



DELINEATION OF THE COMMON SCREW.

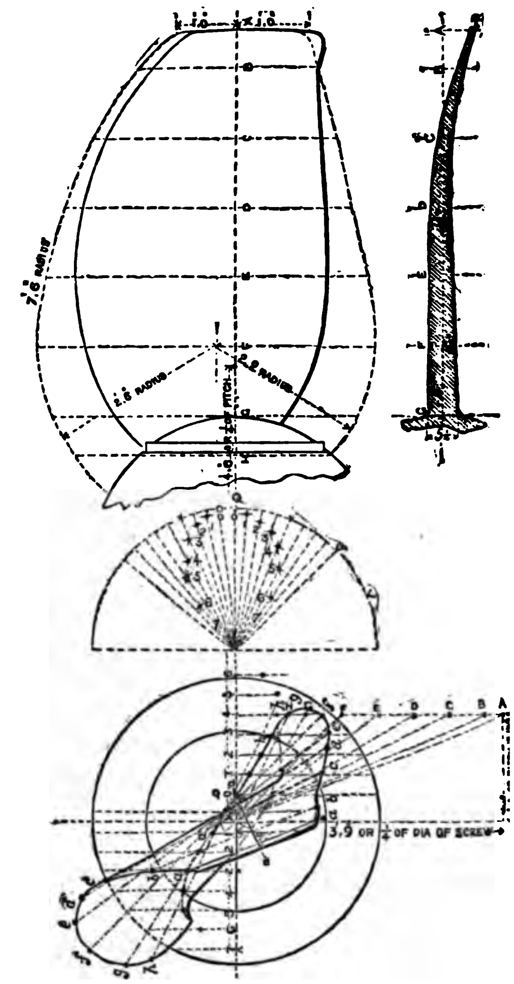
28 feet, the diameter 16 feet, and the length 3 feet. The correct form of the *edge* of the blade is known by *dividing the pitch and circumference into equal divisions*, or by dividing *half* the pitch and circumference into equal parts; and a *continuous curve drawn through the intersections of the horizontal and vertical lines*, denotes the requisition. In the illustrated example, the side, and end elevations, and plan are shown in full lines, to more clearly portray the application of the diagram. The mangin screw propeller is similarly deduced also, by recognizing the variation in the pitch of the leading and following half of the blade.

*** Geometrical Delineation of Griffiths' Screw Propeller.**—The mode of determining the form of this example is fully shown by Fig. 259. As description in this case could not do justice to the principle, it has been considered better to introduce letters and figures of reference corresponding with each view.

* Burgh's "Screw Propulsion."

Water Propulsion.—In page 326, the "prop suitable for the mode of propulsion now under not illustrated by Fig. 201. The principle of this sy

Fig. 259.



Scale ⅜ = 1 *foot.*

DELINEATION OF GRIFFITHS' SCREW PROPELLER.

is, that the water under the hull is permitted to rise through the turbine, and from thence is forced the sides of the ship. Now the power derived b action is readily known by the simple formula; ($-$ R; where C = centrifugal force, G = gravity fluid, and R = resistance. The resistance to be ove is the friction + gravity of the fluid during its tr through the turbine and discharge-pipes; and obvi the higher the ascent of water, the greater the requisite. Now the fall of the water outside th is the main effect, simply because the friction is neutral, and gravity, therefore, in full power; whi

a certain extent, compensates for the friction incurred previously.

The next consideration is the height the water should be raised, to produce the best *effect*, which is only evident by knowing the exact length of the unbroken volume from the outlet at the ship's side proportionate to its sectional area. That this water is of vital importance is obvious to all who consider that a spray cannot be as powerful in its impact as a solid volume of similar contents, due to the interspersion of the *air* between the currents.

Now, according to Professor Rankine, the "gross power of a fall of water" = "the weight of water discharged in a given unity of time into the total head; that is, the vertical elevation of the upper surface of the water at the points where the fall in question begins and ends." The Professor's mode of expressing this is thus:—Let Q = the flow or volume of water discharged in cubic feet per second; D = weight of a cubic foot of water in lbs.; H = the total head: then D Q H = the gross power in foot lbs. per second, which being divided by 550 = the gross horse-powe

As, however, there is a certain loss of *head* power, from the energy being expended at that point, the consideration of the loss is requisite, and the formula by Professor Rankine for this purpose is arranged thus:—$(l - K)$ D Q H = the effective power when K = the loss of the "head" power. The efficient power = $(l - K)$ when KH = the loss of head, and $(l - K)$ H = the effective head.

Proportions of Paddle Wheels.—The rules here given are approximately correct for practical purposes, having been deduced from practical results.

Depth of central immersion for the lowest float from the line of flotation = *draft of hull* ÷ 2 *to* 3.

Diameter of polygon = *depth of the central immersion of the lowest float from the line of flotation* × 5 *to* 7.

Width of float = *depth of the central immersion alluded to* × 1 *to* ·75.

Area of each float = *area of the immersed section of the wheel parallel with the keel of the hull, from the line of flotation to the centre of the float* × ·75 *to* 1.

Pitch of the floats = *width* × 1·5 *to* 2.

Nominal horse-power for each paddle-wheel = *total area of the floats immersed in feet* × 1·3 *to* 2.

These proportions are proportionate to the speed of the wheel and the draft of the hull, being further explained in the Author's "Pocket-book of Rules."

Proportions of the Common Screw Propeller.—Depth of the top of the blade's immersion from the line of flotation = *diameter of screw* ÷ 10 *to* 12.

Pitch of blade's edge = *diameter* × 2 *to* 1·25.

Length of screw = *diameter* ÷ 5 *to* 6.

Proportions of Griffiths' Screw Propeller.—

Diameter of boss = $\dfrac{\textit{diameter of screw}}{3 \textit{ to } 4}$

Length of boss = *diameter of flange* + ½ *diameter of shank, in some cases diameter of boss.*

Diameter of flange of blade = *diameter of boss* × ·5.

Thickness of flange at edge = $\dfrac{\textit{diameter}}{20}$

Lap of blade on boss beyond flange = ¾ *of an inch per foot of diameter of screw.*

Width of blade at widest part = $\dfrac{\textit{diameter of screw}}{3}$

Width of blade at point = $\dfrac{\textit{diameter of screw}}{7}$

Thickness of blade at root = ½ *of an inch to each foot's diameter.*

Thickness at point = ⅛ *of that at root.*

Diameter of shank = *diameter of boss* × ·25.

Metal around shank = $\dfrac{\textit{diameter of boss}}{23 \textit{ to } 24}$

Metal beyond flange and cotter = $\frac{7}{12}$ *of depth of cotter.*

Width of main cotter = *d ameter of shank* × ·5.

Thickness of main cotter = $\dfrac{\textit{diameter of shank}}{6}$

Thickness of feathers in boss = $\dfrac{\textit{diameter of boss}}{40}$

Width of small cotter = $\dfrac{\textit{diameter of boss}}{20}$

Thickness of small cotter = $\dfrac{\textit{width}}{2}$

Angle in side of wedge box = 7½ *degrees.*

Metal in cheeks where cotters enter = $\dfrac{\textit{diameter of boss}}{40}$

Thickness of plate over wedges = $\dfrac{\textit{diameter of boss}}{48}$

Blades of screw to curve forwards ½ inch, to each foot of diameter of screw, from face at root, curve to commence at centre of blades.

Positive Speed and Slip of the Paddle Wheel.—This matter resolves itself into simple calculation in the following manner:—

Speed of floats = *circumference of central line of immersion or polygon* $\times$ *unit of time.*

Slip of wheel or floats = *result of the preceding formula — actual speed of ship.*

Positive Speed and Slip of the Screw Propeller.—

Speed of screw per minute = *pitch of screw* $\times$ *number of revolutions per minute.*

Theoretical speed of ship in knots per hour =

$$\frac{\textit{speed of screw in feet per hour}}{6080 = \textit{Admiralty knot in feet}}$$

Loss of speed or slip of screw = *theoretical speed of ship minus actual speed of ship.*

Actual speed of ship = *speed of screw minus slip.*

To ascertain the actual pitch, required at a given speed of the screw, to produce a given speed of the ship, the rule will be as follows :—

Pitch of screw in feet =

$$\frac{\textit{actual speed of ship in feet per hour}}{\textit{number of revolutions of screw per hour} \times \cdot 9 \textit{ to } \cdot 75}$$

This rule allows a slip or loss of speed of 10 to 25 per cent., 20 per cent. being the average for war ships.

Negative Slip of the Screw Propeller.—The pitch of the propeller multiplied by the number of revolutions in a certain time = its relative advance. Now if the progress of the hull to which the screw is fitted falls short of this sum, the result is termed propeller or positive slip; but if matters are reversed, the application of hull or negative slip results. Now the cause of this latter effect has not been rendered obvious, although various and talented authorities have investigated the subject. The author's ideas on this matter are—that the advance of the speed of the ship over that of the screw is due to the pressure of the water on the after-body of the hull directly in advance of the screw, and that the form of the hull and pitch of the screw determine this result.

In the case of the after lines of the hull being fine, and those forward full, the water displaced forward will incline inwards when the maximum section of the displacement is passed. The principle of the screw's action is, then, to drag the water immediately in front of it; and doubtless with fine lines aft, and a propeller with a coarse pitch, a backward vortex is also caused, and thus the surrounding volume presses against the stern portion of the hull, thereby producing negative slip for the propeller and *positive* slip of the hull This theory has also been proved by the author not to b without foundation in practice. By dropping a piece c wood in front of a revolving propeller at full speed, th wood was forced directly in a line with the ship's progre against the hull; thus proving that a current of wat was in advance of the propeller, proceeding at a great speed than the hull.

A Paper on "Some Remarks on Apparent Negati Slip," by Professor Rankine, was read at the Institutio of Naval Architects on the 11th day of April, 1867, an is here reproduced with the sanction of its author.

"1. When the attempt has been made to account f the apparent negative slip of a screw propeller by th fact of its laying hold of a current of water that following the ship, this objection has been raised: th the forward momentum impressed on that current in second is equivalent to the resistance of the ship; th the backward momentum impressed by the screw c the propeller-race in a second is equivalent to the thru of the screw, which is equal and opposite to the resistan of the ship; and that, consequently, even if the scre were to take hold of every particle of the followin current, that fact would account for a diminution c positive slip only, but not for negative slip.

"2. If the velocity of the following current in whic the screw worked were simply the mean forward velocity of the ship's wake, the objection in question would be unanswerable; for it is the momentum per second due to that mean velocity which is equivalent to the resistance of the ship, and to which the reasoning just mentioned applies.

"3. But the water, affected by the passage of the ship through it, has various reciprocating or wave-like motions combined with the mean velocity of the wake; and, in particular, there is forward motion under every crest, and backward motion under every hollow, of the waves that accompany the ship. The velocity of those reciprocating motions is not connected directly with the resistance of the vessel—in fact, their resultant momentum is equal to nothing; and it is only the momentum of the uniform current, which remains after the wave-motions have died out, that is equivalent to the ship's resistance.

"4. Hence, if there happens to be, as there generall is, the crest of a following or filling wave under th ship's counter, the water of which the screw lays hol has a temporary forward velocity over and above th permanent velocity of the wake; that temporary fo

ward velocity, indeed, may be many times greater than the permanent velocity of that current whose momentum is equivalent to the resistance of the ship ; and thus any extent of apparent negative slip may be accounted for.

"5. The existence of a following wave explains also the fact that any considerable apparent negative slip is always accompanied by waste of motive power, the resistance to the motion of the engine increasing in a greater proportion than its speed is diminished. For amongst the laws of wave motion are the following: that all forward motion of the particles in a wave is accompanied by an elevation of level, and that the pressure against a body in front of the wave, due to that elevation of level, is exactly equal to the pressure required to impress the forward motion upon the particles of water. Such is the pressure exerted upon the stern of a ship by the wave which follows under her counter, when that wave is undisturbed by the action of the screw. But the screw, by checking or reversing the motion of the particles of water, lowers the level of the crest of the following wave, and diminishes the forward pressure which that wave exerts on the vessel. That diminution of pressure is virtually equivalent to an increase of the ship's resistance; so that the thrust of the screw must be equal not merely to the resistance properly due to the dimensions and figure of the ship, but to that resistance increased by a force equal to the diminution which the action of the screw produces in the pressure exerted on the ship by the following wave. Thus the total thrust of the screw is increased above its effective thrust—that is, above the proper resistance of the ship, in a proportion greater than the proportion in which the speed of the screw is diminished through apparent negative slip, so that the result is an increased expenditure of motive power above what would be required if the screw acted in water not affected by wave motion.

"6. The principles of the preceding paragraph do not apply to uniform forward motion of the particles of water produced by friction, because such motion is not accompanied by the production of a swell, and hence the permanent following current in the ship's wake due to frictional resistance does not give rise to a loss of thrust as the wave motion of the particles of water does."

THE PRINCIPLES OF THE MARINE BOILER.*

Priming.—This phenomenon is the terror of the engineer and stoker, inasmuch that it occurs without warning, and as suddenly ceases. The stoker may turn his back for a moment or two, and on viewing the water gauge on his return, may find it empty, instead of half full as before. The work of an instant is to open the furnace doors, close the dampers, and either damp or draw the fires, while the engineer has put on the feed from the donkey. Apart from the danger of explosion and burning the boiler, "priming" impedes the progress of the engines, and in some instances bursts the covers and ends of the cylinders when the relief valves are insufficient to release the water in time to prevent fracture.

Now, as to the causes for priming, they are mostly, change of the feed water, smallness of the steam space in the boiler, lowness of the steam space above the water line in the boiler, the non-congregation of the steam, or the exit of the steam exceeding the supply—being the result of opening the stop valves too much, incorrect proportion of the heating surfaces, and the arrangement internally. Obviously, then, if these causes are known, the engineers, designing on land and in charge at sea, should not fail to obviate them as far as practicable.

Feed Pump.—The cubical contents of this detail bears a relation to the pressure of the steam and capacity of the cylinder and steam passages formed therewith. The formula is readily attained by recognising these relations as follows:—*Cubic contents of cylinder and steam passage in feet* $\times$ 3 *times the number of cubic inches of water to produce* 1 *cubic foot of steam according to the maximum pressure required = cubic contents of the feed pump in inches for one engine.*

The table of the relative volumes of the steam and water being of importance, it is therefore introduced:—

Pressure of Steam in lbs. per square inch.	Cubic Inches of Water to 1 cubic foot of Steam.
10	1·7
15	2·0
20	2·3
25	2·6
30	2·9
35	3·2
40	3·5
45	3·8

* See Burgh's "Boilers and Boiler-making."

Pressure of Steam in lbs. per square inch.	Cubic inches of Water to 1 cubic foot of Steam.
50	4·0
60	4·6
70	5·1
80	5·65
90	6·2
100	6·68

Cubic Contents of Steam Space = *four to six cubic feet per superficial foot of grate surface,*

Area of Grate Surface in Square Feet.—This proportion depends on the quality of the fuel employed, and amount of steam required proportionate to the *time* for combustion; also the pressure of the steam must not be neglected, and its traverse from the boiler to the engine. As in practice the combustion is retarded by the slag or clinker forming, it is essential to retain a maximum area, which under adverse circumstances will be sufficient. An exposition of the principles of combustion is afforded in detail in Chapter III.; and, therefore, attention must be directed to it before the matter in question can be thoroughly appreciated. The proportion of the indicated to the nominal horse power must also be noticed, so that if an engineer desires to attain a high ratio, the means of development must be produced in the form of fuel and water to cause the steam.

The practice of the engineers of the present day is to make the area in question = *horse power nominal collectively* × ·68 to ·75 when the indicated power = nominal power × 5, and the consumption of the fuel about 2½ to 4 lbs. per indicated horse-power per hour on the voyage.

Area of Tubular Passage = *area of fire grate* ÷ 6·5 to 7: this is the general practice for construction, but doubtless the effective area is not more than ⅛th when the lower tubes are choked.

Area of Chimney = *area of tubular passage* × ·6 to ·75 when the height = 6·5 diameters, which is the usual proportion.

Area of Safety Valve in Square Inches.—This area must bear a strict proportion to the fire grate, as the utility of the former is due to the effect of the latter: a good and universal rule = *grate surface in feet* ÷ 3, which is the practice in the present day, recognising all the essentialities.

Superficial Areas of Heating Surfaces.—These proportions are fully investigated in Chapter III., both explanatorially and tabular; from which the best practice is evident, and the proportions can be deduced by simple division.

Construction and Strength of Boilers.—The strains imposed are tensile, compression, shearing, and torsion; the latter in an indirect form. Mr. Fairbairn, in his work already alluded to, states, when alluding to the "strength of plates":—

Comparative results of rolled iron as derived from experiment, the Yorkshire plates being unity.

Names of iron.	No. of Experiments.	Mean Breaking Weight in Tons per square inch.	Mean Breaking Weight in Tons per square inch.	Ratio of the Strength of Plates drawn in the direction of the Fibre, and across it. Also of rolled and faggoted Bars drawn in the direction of the Fibre.
Yorkshire plates .	8	25·514	..	..
Derbyshire plates .	4	..	20·160	1 : 0·7882
Shropshire plates .	4	..	22·413	1 : 0·8789
Staffordshire plates	4	..	20·264	1 : 0·7946
Mean . . .	..	25·514	20·945	1 : 0·8209
From Mr. Telford and Capt. Brown's experiments on bars	..	..	26·41	1 : 1·0351

When treating of the strength of "riveted joints," Mr. Fairbairn's conclusions are:—

Cohesive strength of Plates, Breaking Weight in lbs. per square inch.	Strength of Single riveted Joints of equal Section to the Plates, taken through the Line of Rivets. Breaking Weight in lbs. per square inch.	Strength of Double-riveted Joints of equal Section to the Plates, taken through the Line of Rivets. Breaking Weight in lbs. per square inch.
57,724	45,743	52,352
61,579	36,606	48,821
58,322	43,141	58,286
50,983	43,515	54,594
51,130	40,249	53,879
49,281	44,715	53,879
43,805	37,161	..
47,062	..	..
Mean 52,486	41,590	53,635

The relative strengths will therefore be:—

For the plate . . .	1000
Double-riveted joint . .	1021
Single-riveted joint . .	791

From the preceding results, it will be seen that the single-riveted joints have lost one-fifth of the actual strength of the plates, whilst the double-riveted have retained their resisting powers unimpaired. These are important and convincing proofs of the superior value of the double joint; and in all cases where strength is required this description of joint should never be omitted.

The best proportions, according to the same authority, are:—

Table exhibiting the strongest forms and best proportions of riveted joints as deduced from the experiments and actual practice.

Thickness of Plates in Inches.	Diameter of Rivets in Inches.	Length of Rivets from the Head in Inches.	Distance of Rivets from Centre to Centre in Inches.	Quantity of Lap in Single Joints in Inches.	Quantity of Lap in Double-riveted Joints in Inches.
·19 = $\frac{3}{16}$	·38	·88	1·25	1·25	
·25 = $\frac{4}{16}$	·50	1·13	1·50 } 6	1·50 } 6	
·31 = $\frac{5}{16}$	·63 } 2	1·38	1·63	1·88	For the double-riveted joint, add two-thirds of the depth of the single lap.
·38 = $\frac{6}{16}$	·75	1·63 } 4·5	1·75 } 5	2·00 5·5	
·50 = $\frac{8}{16}$	·81	2·25	2·00	2·25	
·63 = $\frac{10}{16}$	·94 } 1·5	2·75	2·50 } 4	2·75 } 4·5	
·75 = $\frac{12}{16}$	1·13	3·25	3·00	3·25	

The figures 2, 1·5, 4·5, 6, 5, &c., in the preceding table are multipliers for the diameter, length, and distance of rivets, also for the quantity of lap allowed for the single and double joints. These multipliers may be considered as proportionals of the thicknesses of the plates to the diameter, length, distance of rivets, &c. For example, suppose we take three-eighth plates, and required the proportionate parts of the strongest form of joint, it will be—

·375 × 2 = ·750 diameter of rivet, $\frac{3}{4}$ inch.
·375 × $4\frac{1}{2}$ = 1·688 length of rivot, $1\frac{3}{4}$ inch.
·375 × 5 = 1·875 distance between rivets, $1\frac{3}{4}$ inch.
·375 × $5\frac{1}{2}$ = 2·063 quantity of lap, 2 inches.
·375 × $5\frac{1}{2}$ = 3·438 quantity of lap for double joints, $3\frac{1}{2}$ in.

·75, 1·68, 1·87, 2·06, and 3·43 are, therefore, the proportionate quantities necessary to form the strongest steam- or water-tight joints on plates three-eighths of an inch thick.

The formula for "stays" is a simple matter, and can be deduced as follows. Let

x = area of stay in square inches.
A = area of surface in square inches supported by the stay.
P = pressure of steam in pounds per square inch.
S = the strain to be resisted.
F = factor of safety in proportion to the strain.

Then

$$x = \frac{A \times P}{S} \times F.$$

Mr. Fairbairn's conclusion as to the material to be employed for "stays" are—

"It will be found that the iron stay and copper plate (not riveted) have little more than one-half the strength of those where both are of iron; that iron stays screwed and riveted into iron plates are to iron stays screwed and riveted into copper plates as 1000 : 856; and that copper stays screwed and riveted into copper plates of the same dimensions have only about one-half the strength of those where both the stays and plates are of iron. These are facts in connection with the construction of locomotive, marine, and other descriptions of boilers having flat surfaces, which may safely be relied upon, and that more particularly when exposed to severe strain, or the elastic force of high-pressure steam."

In arranging the "stays" retain a distance of sixteen inches between them above the fire-boxes, and twelve inches pitch, below the tubes: a fair specimen of this matter is represented in plate 35.

Superheater.—This apparatus has been illustrated in detail, and as the type determines the proportion, a fixed ratio is not available. For tubular superheaters the superficial area in feet = *nominal horse power* × 1·5 *to* 2; with steam at 80 to 120 lbs. on the square inch, while in some instances 2·5 to 3 square feet per horse-power are adopted for lower pressures

STATISTICAL PROPORTIONS of the WEIGHTS of ENGINES, BOILERS, WATER, SPARE GEAR, &c. &c.

TYPE OF ENGINE.	Weight in Cwts. per Nominal Horse-power.								Cubical Contents of Bunkers in feet.	Number of Days' Steaming.
	Engines.	Boilers.	Water in Boilers.	Propeller and Shafting.	Spare Gear.	Coal Bunkers.	Coal.	Total, including Fittings.		
Trunk	3 to 2·5	4 to 4·3	2·5 to 2·7	1·75 to 2	·7 to ·75	·4 to ·5	13·5	12 to 16	31	4·5 to 6·5 Full Steam. 6 to 8 days expansively.
Direct Acting . . .	4·3 to 4·8	4·5 to 5	2·5 to 2·8	2 to 2·1	·8 to 1	·56 to ·7	13·6	15·6 to 18	32·125	
Return Action . . .	4·5 to 5	5 to 5·6	2·5 to 3	2·25 to 2·3	·98 to 1·1	·68 to ·8	14	17 to 20	33 to 35	

RETURN of TONNAGE, POWER (Nominal and Indicated), DESCRIPTION of ENGINES, and DISPLACEMENT of the Ships "OCTAVIA," "CONSTANC and "ARETHUSA," setting forth the RESULTS of the TRIAL from PLYMOUTH to MADEIRA, in a Tabulated Form, showing the Time Departure and Arrival; when previously Docked; the Total and Daily Number of Hours under Steam; Total and Daily Distan Steamed; Total and Daily Consumption of Coal; Description and Quantity of Coal put on Board; the greatest Number of Revolutio on any Day; Distance run, with the Average Pressure on that Day; and if prevented at any Time from Steaming, the Cause and Peri of Stoppage.

"OCTAVIA."					"CONSTANCE."					"ARETHUSA."				
Tonnage.	Horse Power. Nominal.	Horse Power. Indicated.	Description of Engines.	Displacement.	Tonnage.	Horse Power. Nominal.	Horse Power. Indicated.	Description of Engines.	Displacement.	Tonnage.	Horse Power. Nominal.	Horse Power. Indicated.	Description of Engines.	Displ[acement]
3,161	500	1,839·1 greatest on passage. 1,399·8 mean on passage.	Horizontal direct with double piston-rods, three cylinders, and surface condensers.	3,747	3,213	500	2,322·3 greatest on passage. 1,747 mean on passage.	Engines on Woolf's principle, direct acting, with inclined cylinders, and surface condensers.	3,669	3,141	500	1,881·9 greatest on passage. 1,052·2 mean on passage.	Direct horizontal trunk, with surface condensers.	3,[illegible]

RESULTS of the TRIAL from PLYMOUTH to MADEIRA.—(Taken from Reports received from the Ships.)

NAME OF SHIP.	Time of Departure from Plymouth.	Time of Arrival at Funchal.	When previously Docked.	Day of Month.	Number of Hours under Steam Daily.	Distance Steamed Daily.	Coal. Consumption Daily.	Coal. Description.*	Coal. Quantity put on Board.	Greatest Number of Revolutions on any Day.	Distance run on that Day.	Average Pressure on that Day.	Cause and Period o[f] Stoppage.
				1865.	Hours.	Knots.	Tons.		Tons.		Knots.	Lbs.	
"OCTAVIA"	6 p.m. 30 Sept. 1865.	6·45 a.m. 9 Oct. 1865.	12 June 1865.	30 Sept.	6	58·1	9·15	Llangennech and Dunraven mixed, good. Carr's Hartley and Welsh mixed, good. Dunraven Merthyr and Thomas Merthyr mixed, inferior.	300	55·75 on 2 Oct. 1865.	211·6	12·78	30th Sept., stopp[ed] for 10 minutes; cau[se] priming of boiler 3rd Oct., stopped 2 hr[s] 52 min.; cause, heatin[g] of centre crank bearing and to shift brasses 4th Oct., stopped 5 hrs 30 min.; cause, shifting brasses of centre crank bearing. 6th Oct., engines stopped at 10·5 p.m., only 15 tons of coal remaining.
				1 Oct.	24	226·5	43·05						
				2 „	24	211·6	51·64						
				3 „	21	152·7	51·15						
				4 „	19	119·2	40·89						
				5 „	24	178·6	47·47						
				6 „	22	105·0	33·39						
				Total . .	140	1051·7	276·74						
"CONSTANCE"	6 p.m. 30 Sept. 1865.	3 p.m. 7 Oct. 1865.	13 June, 1865.	30 Sept.	6	68·85	12·2	Mixture of Dunraven Merthyr, Cameron, Coalbrook, and Gellia Cadoxton, good.	277	53·9 on 1 Oct. 1865.	255·95	21·13 high pressure. 4·15 low pressure.	4th Oct., stopped 10 hrs.; cause, heavy se[a] and head swell. 6th Oct. 8·50 a.m., eased engines in consequence of heavy head sea an[d] westerly gale.
				1 Oct.	24	255·95	50·3						
				2 „	24	219·65	47·2						
				3 „	24	188·40	48·0						
				4 „	14	103·85	26·7						
				5 „	24	192·6	44·2						
				6 „	8	61·4	13·9						
				Total . .	124	1090·7	242·5						
"ARETHUSA"	6 p.m. 30 Sept. 1865.	5·35 p.m. 10 Oct. 1865.	20 Sept. 1865.	30 Sept.	6	67·0	20·925	Llanelly, Llangennech, Lambton's Wallsend, fair.	260	60·5 on 1 Oct. 1865.	255·6	12·76	1st Oct., stopp[ed] 10 min.; cause, g[ud]geon-end of after c[on]necting rod heat[ed]. 6th October, engi[nes] stopped at 7·15 a[.m.], only 31 tons of coal [re]maining.
				1 Oct.	24	255·6	59·675						
				2 „	24	223·8	47·7						
				3 „	24	163·28	50·5						
				4 „	24	95·76	22·55						
				5 „	24	164·08	21·85						
				6 „	8	61·0	5·65						
				Total . .	134	1030·52	228·85						

* The coals used were mixed, in the proportion of one-third North Country to two-thirds Welsh.

Department of the Controller of the Navy, 20th March, 1866.

ROBERT SPENCER ROBINSON.

THE DUTIES OF ENGINEERS IN CHARGE AFLOAT.

Before Starting.—Over-haul condensers, pumps, and valves, test nuts of bearings; while the stokers have filled the boilers and lighted the fires; allow the safety valves to be open until steam blows.

Before Leaving Harbour.—Steam being up, before moving engines, open all the stop valves, feed, &c., fill lubricators, attend to worsteds, see all clear, blow through cylinder and condensers, try engines ahead and astern, before stating ready to start.

Order in Stoke-hole.—The amount of steam pressure required without variation; stoke freely when under steam, but not too heavily, so that when stopping suddenly, the combustion can be lessened by opening the flue or smoke box door and closing the damper doors, and thus the rate of combustion is reduced.

Before Engineer takes Charge.—The one leaving must report all right; and in the event of heated bearings, the one in charge should stop till all is cool.

On Change of Watch.—Engineer in charge examines all bearings, fills lubricators, and inspects fires; in case of cleaning fires, see that the fires are burnt enough for cleaning, and complete his log, which is done hourly.

Engineer on Duty.—Learn pressure of steam in general required; see that the level of water is correct as indicated by the gauge glass and cocks, blowing through to test them. Examine all bearings before taking charge, sound the bilges, or take depth of the water in the engine room and stoking-hole bilges; test the density of the water in the boiler by the salinometer.

In the Case of Stopping Suddenly.—Open safety valves, particularly when the fires are brisk; when captain gives notice when going in harbour, fire light when he requires no more steam. The relief valves should be opened after stopping a quarter of an hour.

In the Case of Heated Crank Pins and Shaft Bearings.—When the symptoms occur, reduce the supply of the steam, turn on the "spray" and "jet" gently, and if the temperature increases, increase the flow of the water: should the latter not be adequate to the occasion, slacken the nuts half a turn at the most, for if the "caps" are loose, a "play" results, injurious to the motion of the engine. Slacken the speed of the engine or stop.

In the Case of the Boilers Priming.—Partially close the supply steam valve; open fire doors, close dampers, turn on feed; and if these emergencies are not sufficient, damp or draw fires; shut steam valves; which latter operations are the last resource.

Regulation of the Feed Water.—Having arranged the proportion of the sea to the condensed steam—if surface condensation—control the supply into the boilers to be proportionate to the formation and expenditure of the steam as far as practicable; but with injection condensers supply the boilers from the discharge-pipe, or hot-well; and (surface) "blow-out" *continuously* to obviate excessive incrustation.

Before Finally Drawing Fire or Banking.—See that there is sufficient water in the boiler for a short stop in port. When a long stay is contemplated, or at the end of the voyage, draw fires and blow out water; finally, take out worsteds, open relief valves, close sea cocks after all is clear, blow through all boiler cocks and gauge, and wipe engines throughout.

CHAPTER X.

DESCRIPTION OF PLATES.

Plates 1 and 2.—These plates are described by the firm as follows:—

The engines are of 300 horse-power, nominal power, manufactured by Messrs. Maudslay, Sons, and Field, having three horizontal cylinders, 55 inches in diameter, with a stroke of 2 feet 6 inches, giving motion to a three-throw crank shaft, with the cranks set at an angle of 120° with each other. The cylinders are entirely cased with steam, having steam jackets around their circumferences and at their ends, and are intended to work with a pressure of 25 lbs. per square inch. The steam is cut off by the slide valve at from ¼ to ⅛ of the stroke, without the use of any separate expansion valves. The slides are double ported as far as the eduction passages are concerned, but they have three steam openings in order to give the requisite area to the admission of the steam. The slides are worked by a crank shaft running across the whole length of the engines, and driven by a train of spur wheels from the main crank shaft of the engines. The two intermediate wheels of this train are mounted in a movable frame, which may be raised or lowered at pleasure, the effect of which is to vary the position of the cranks that work the slides, so as to make them cut off sooner or later, as may be required: by the same means the slide cranks are shifted into the proper position for going astern. There are two surface condensers, fitted with inclined tubes, with the steam inside the tubes, and the condensing water outside, as originally adopted by Samuel Hall. The steam passes into the condenser at the upper end of the tubes, and the condensed water is delivered at the lower end, from whence it passes to the air-pumps. Each condenser is fitted with 2,436 copper tubes, 6 feet 3 inches long, ½-inch diameter inside, with a thickness of No. 18 wire-gauge. The tubes are packed at each end with small screwed stuffing boxes with tape packings. The condensers, with their corresponding air-pumps, are placed opposite the two outer cylinders, the air-pumps being worked by brackets secured to the piston rods. The space between the two condensers over the cylinder cross-heads and piston rods is occupied by the engine platform; by this arrangement all the working parts are kept well within the reach of the person in charge. Motion is imparted to the circulating pumps by rods, connected with the steam pistons; these pumps are 10½ inches diameter, and 2 feet 6 inches stroke, and are fitted with lignum-vitæ packing to the plungers, and india-rubber foot and discharge valves. The condensed water is delivered from the air-pumps into a feed-water tank, from which it is forced into the boiler by the ordinary feed pumps. This description of engine offers the advantage of great economy of fuel, with a comparatively simple arrangement, while at the same time it gives an equable power, and the centre of gravity of the different moving parts being always the same in relation to the vessel, it does away with all the unpleasant vibration so frequently felt with quick-working engines.

The steam is generated in two boilers, of the ordinary tubular construction with return tubes over the furnaces, placed one on either side of the vessel, with the stoker's room between them. Each boiler has four furnaces, 3 feet 2 inches wide, and 6 feet 7 inches long, and is fitted with 440 brass tubes, 2½ inches diameter outside, and 6 feet long; the chimney is 5 feet 6 inches diameter, and it is made to lower on the telescope principle.

Each of the boilers is fitted with a superheating apparatus, consisting of a series of flattened tubes, placed in rows in the up-take: the tubes are placed so as to offer as little obstruction as possible to the heated air in its passage to the chimney; each superheater has 81 tubes, 6 feet 6 inches long, and 2½ inches diameter before they are flattened.

The screw-propeller is four-bladed, with an increasing pitch on Mr. Woodcroft's principle; it is 12 feet diameter, and has a mean pitch of 13 feet. The blades are bolted on to a spherical boss, and they are made so that the pitch can be altered and set at any pitch, from 11 feet 6 inches to 14 feet 6 inches.

The vessel is an iron-plated battery ship, with a ram

bow, and was built by the Thames Iron Works and Shipbuilding Company; it is 2,811 tons, builders' measurement, and is 220 feet long between perpendiculars, and 53 feet wide, with a draught of water of 14 feet 6 inches.

Plates 3 to 8 inclusive.—These illustrate the arrangement and details of Mr. Spencer's patent auxiliary surface condensers and engines for the iron screw steam-ship "Frankfort," of Liverpool. The type of the engine is vertical, inverted direct action. The condensers are tubular, with outside condensation; the pumps being on the combined principle, similar to those adopted by Messrs. Penn in H.M.S. "Arethusa." The motion is derived from the steam piston-rods for the pumps, and the arrangements and form of the valves can be understood from the drawings. The cylinders are fitted with the ordinary slide valves and expansion valves at the back of the same.

The following particulars will render the several portions to be understood without further notice :—

Diameter of Cylinders	20″
Length of Stroke of Piston	15″
Diameter of Combined Pumps	14″
Length of Stroke	15″
Diameter of Feed Pumps	$3\frac{1}{2}$″
Length of Stroke of ditto	15″
No. of Tubes in Condenser	584
Length of ditto	5′ 6″
External diameter of Tubes	$1\frac{1}{4}$″
Thickness of ditto, 18 wire gauge.	
Amount of Cooling Surface (sq. feet), 1,004.	

Plates 9 to 14 inclusive.—A description of these engines, constructed by Messrs. Dudgeon, will be found in pages 71 to 73, inclusive. The "condensers" are described in page 259; also the "cylinders" in page 277; the "slide valve" in page 286; the "piston" in page 278, and the "link motion" in page 292; to which the following particulars, supplied by the firm, are now added :—

Nominal Horse-power Collectively	350
Actual „ Indicated	1540
Diameter of Cylinders	Inner, 27″; Outer, 62″
Length of Stroke	2′ 0″
Width of Exhaust Steam Ports in Cylinders,	$6\frac{3}{4}$″ and 4″
„ „ „ Valve	$3\frac{1}{4}$″
„ Supply Ports in Cylinders	4″ and 2″
„ „ Valve	$3\frac{1}{4}$″
„ Small Bar in Cylinders	$1\frac{1}{2}$″
Width of Large Bar in Cylinders	$4\frac{1}{2}$″
Length of Steam Ports	2′ 6″
Outside Lap of Valve	$1\frac{7}{16}$″
Half travel of Valve	$3\frac{1}{8}$″
Width of Steam Opening caused by Valve	$1\frac{11}{16}$″
Diameter of Valve Rod	$3\frac{1}{4}$″
Length of Link between Centres of Eccentric Rods	1′ 6″
Thickness of Cylinders	Inner $1\frac{1}{8}$″; Outer $1\frac{1}{16}$″
Diameter of Piston Rods	Inner Cylinder (1) $3\frac{7}{8}$″; Outer (2) $4\frac{1}{8}$″
Depth of Piston	8″
Diameter of Crank Shaft	(bearing) $9\frac{5}{8}$″
Length of Bearings	$18\frac{3}{4}$″
Diameter of Connecting Rod	$9\frac{5}{8}$″
Length of „ „	4′ 0″
Diameter of Air Pump	11″
„ Suction Valves' Openings	$4\frac{3}{8}$″
„ Discharge „ „	$4\frac{3}{8}$″
Number of Suction Valves (ditto for each condenser)	18
„ Discharge „ .. .. (ditto)	18
Diameter of Circulating Pump	11″
„ Suction Valves Openings	As above.
„ Discharge „ „	Ditto.
Number of Suction „ „	Ditto.
„ Discharge „ „	Ditto.
Thickness of Metal of Condenser	$\frac{11}{16}$″
Number of Tubes in Condenser	2017
Diameter of Tubes (External)	$\frac{5}{8}$″
Length of Tubes	5′ $10\frac{1}{4}$″
Diameter of Feed Pumps	$3\frac{3}{4}$″
„ Injection Pipe (Surface Condenser, with additional Jet Injection 3″ diameter.)	
„ Feed	3″
„ Bilge Injection Cocks	3″
„ Cylinder Relief Valves	4″
„ Main Discharge Pipe	12″
„ Exhaust Steam	13″
„ Single Supply	$9\frac{1}{2}$″
„ Main Supply	16″
Material of Air Pump	Gun Metal.
„ Feed Plunger	Ditto.
Feed Pump	Single Acting.
Distance between Centres of Cylinders	5′ 6″
Diameter of Screw Propeller	10′ 6″
Length on Line of Keel	3′ 0″
Pitch	18′ 0″
Kind of Propeller	Common.
Number of Blades	3

Length of Ship	288 feet.
Breadth	34 feet.
Depth	27 feet.
Speed	13 knots per hour.
Number of Revolutions of Engines	90 per minute.
Kind of Boilers	Multitubular.
Number of Boilers	Two.
Length of Tubes in each, 6′ 3″; and total number	608
„ Fire Grates	6′ 6″; Breadth 3′ 1¾″
Number	Ten.
Diameter of Funnel	4′ 6″
Consumption of Fuel per indicated Horse-power,	2·667 lbs. per hour.

Plates 15 to 18 inclusive.—These engines are constructed by Messrs. Napier, and fitted by them in H.M.S. "Hector," described in pages 46 to 49 inclusive. The firm have added the following dimensions and particulars:—

Nominal Horse-power Collectively	800
Diameter of Cylinder	82″
Width of Exhaust Steam Port in Cylinder	8½″
„ „ „ Valve	12″
„ Supply Port in Cylinder	3″
„ Small Bar in Cylinder	1¾″
„ Large „ „	12″
Length of Steam Ports	55″
Outside Lap of Valve	3½″ and 3⅞″
Half travel of Valve	5½″
Diameter of Valve Rod	4½″
Length of Link between centres of Eccentric Rods	24½″
Diameter of Piston Rods	6½″; 10½″ in body
Depth of Piston	14″ on edge
Length of Stroke	4′
Diameter of Crank Shaft	16½″
Length of bearings ditto	(total) 9′ 1½″
Diameter of Connecting Rod	(small end) 8½″
Length of ditto	(centres) 8′ 9″
Diameter of Air Pump	(single acting) 43″
„ Suction Valves' Openings	6½″
„ Discharge ditto	9⅛″
No. of Suction	(for each pump) 42
„ Discharge	„ „ 24
Diameter of Feed Pump	6½″
„ Injection Pipe (2)	7½″
„ Feed	(main) 6″
„ Snifting Valve	4″
„ Bilge Injection Cocks	4½″
„ Cylinder Relief Valves	6½″

Blow-through Valve (slide valve of starting cylinder)	
Diameter of Main Discharge Pipe	(2)
„ Exhaust Steam „	(2)
„ Single Supply Steam Pipe (to 1 engine)	
„ Main „ „ „	
Thickness of Cylinders	
„ Condenser	
Material of Air Pump	B
„ Feed Plunger	D
Feed Pump	Single Act
Distance between centres of Cylinders	8
Diameter of Screw Propeller	
Length on line of Keel	4′ 4½″ b
Pitch	
Kind of Propeller	Griffi
No. of Blades	
Length of Ship	280′
Breadth	56′
Depth	38′
Kind of Expansion Gear. Eccentric motion and brating cylindrical valves.	
Grade of Expansion, maximum	
„ „ minimum	
Kind of Boilers	Tubul
No. of ditto	
Length of Tubes in each, 6′ 6″, and total number, 3,168; diameter	
Length of Fire Grates, 7′ 5″, breadth	3′
No „ „	
Diameter of Funnel	9′

Plates 19 to 21 inclusive.—In pages 62 to 65 clusive there will be seen a description of these engir constructed by Messrs. Maudslay. A detailed notice the "condenser" in page 250; of the "cylinders" page 276; of the "slide valves" in page 287; and link motion in page 289. The firm have also contribu the following particulars:—

The engines are of 900 horse-power (nominal). Th are horizontal, double piston-rod engines, having t cylinders 87 inches in diameter, with a stroke of 4 fe and intended to work at about 60 revolutions minute. The cylinders are entirely cased with stea having steam jackets round their circumference and their ends, and are intended to work with steam at pressure of 20 lbs. per square inch. The slide valv are double ported, the steam ports in the cylind being 68 inches long by 3¼ inches wide. The expansi

valves are revolving valves with four ports, and revolve at one-half the speed of the engines; the degree of expansion is regulated by spiral grooves cut in long brass sockets, as introduced and practised for some time by Maudslay, Sons, and Field. The air-pumps are horizontal, double acting pumps $26\frac{3}{4}$ inches in diameter, worked by rods direct from the pistons. The foot and discharge valves are of india-rubber with gun-metal seatings; they are placed above the level of the air-pump, the foot valves being inverted, and draining directly from the bottom of the condensers, which are placed altogether above the level of the air-pumps. The feed-pumps are worked from the cylinder crosshead, the pumps being placed under the starting platform at the end of the condenser; the valve boxes of the feed and bilge pumps are placed so as to be easily got at at all times when the engines are at work.

The screw propeller is four-bladed with moveable blades, so arranged as to allow of the pitch being readily adjusted. It is 20 feet in diameter. and the pitch can be adjusted from 20 feet to 25 feet.

The boilers are in six parts, placed forward of the engines. They have in all 30 fire-places, 7 feet 6 inches long by 3 feet wide, having an area of firegrate of 675 square feet, and are fitted with 3,600 brass tubes $2\frac{1}{4}$ inches in diameter outside and 6 feet 6 inches long; the total area of heating surface is 19,000 square feet. The two aft boilers are fitted with a superheating apparatus consisting of a number of flattened tubes placed in the boiler up-takes immediately above the level of the boiler tubes. There is one chimney 9 feet 4 inches in diameter, which is made to lower on the telescopic principle.

Plate 22.—Description of the engines and boilers of the Egyptian screw ships "Charkieh" and "Dakahlieh," constructed by Messrs. J. and G. Rennie, with surface condensers and superheaters by the firm:—

These engines are on the return connecting-rod principle, with double piston-rods.

The two cylinders are on the same side of the ship and bolted together, leaving a passage between them to get at the centre bearings.

Each cylinder is $63\frac{3}{4}$ inches in diameter and 3 feet stroke.

The slides are double ported, and a separate valve (gridiron) is supplied for working expansively.

The surface condenser, on the opposite side of the ship, is fitted with vertical copper tubes $\frac{9}{16}$ inch diameter.

The steam passes through the tubes; each tube is fitted with a screw gland and stuffing-box top and bottom, and cotton packings.

Total condenser surface about 7,000 square feet.

Circulating and Air Pumps.—One of each: both are worked from the steam pistons, and are of equal diameter, and the passages are so arranged that a circulating pump can also be worked as an air-pump if the condensers are worked with injection.

The water is drawn through the condensers, and not pushed through by the circulating pump, so as to cause a more uniform flow, and dispense with shocks in the condenser. The quantity of cold water circulated is in proportion to the quantity of steam used. The air pump discharges the condensed steam into a hotwell or overboard.

Feed Pumps.—Each engine is fitted with a full-sized feed-pump in case of working by injection, drawing the water either from the hotwell or sea.

An injection valve and pipe, of full size for working with ordinary condensation, is fitted, as well as a small injection pipe for working up the quantity of water lost by evaporation.

Bilge Pump.—Each engine is fitted with one of same size as the feed-pump.

Boilers.—Four in number, having a total heating surface of about 7,500 square feet, and grate surface of 280 square feet, with Lamb's patent flues in place of tubes. Each boiler is fitted with a brine or blow-off pipe, feed regulating valve, double safety valves, stop valves, and vacuum valves.

The boilers are united into one take-up, into which is fitted a Lamb's patent Scroll Superheater, of about 800 square feet of surface, so arranged that engines may be worked with or without superheated steam.

Shafting—The bearings in the sea are fitted with lignum-vitæ.

Screw.—Three-bladed, 15 feet 6 inches diameter, and 19 feet 6 inches pitch.

The dimensions of the vessels "Charkieh" and "Dakahlieh" are as follows:—

Length between perpendiculars	260 feet
Beam	35 „
Depth amidships	26 feet 6 inches
Tonnage, O M	$1557\frac{25}{94}$

The trial of the "Charkieh" gave the following results:—

Draft of water, forward	15 feet 6 inches
Ditto aft	17 „ 9 „
Displacement at 16 feet 7½ inches ..	2,200 tons
Midship section	462 square feet

Engines.—Horizontal double piston rod. Nominal horse-power, 350.

Screw, 6-bladed, with wrought-iron blades—

Diameter	15 feet 9 inches
Pitch	19 „ 6 „

Engines with surface condensers and superheaters.

Trials outside Plymouth Breakwater:—

Revns.	Steam.	Vacuum.		Min.	Sec.		Knots per Hour.
74	23	24½	First Run ..	4	52	=	12·329
74	24	24	Second do ..	4	13	=	14·229
76	23	25	Third do. ..	4	52	=	12·329
74	22	25	Fourth do...	4	27	=	13·483
						4)	52·370

Mean of Four Runs	13·0925
Ditto, by Government Rule	13·185

	Days.	Hrs.	Miles.
Plymouth to Gibraltar	3	18	1,080
Gibraltar to Malta	3	18	981
Malta to Alexandra	3	0	819
	10	12	2,880

Mean speed, Plymouth to Alexandria .. 11·4 knots.

When working *at sea*, with 18½ lbs. steam, 71 revolutions, 24 vacuum, 18½ lbs. pressure, the result was 1,475 indicated HP.

When cutting at ⅓-stroke, 58 revolutions, 18½ lbs. steam, 24 vacuum, indicated power at sea, 1,166.

When working on the first trial trip, with 3-bladed screw, before 6-bladed screw was put on, indicated power, 1,860 horses; pitch of screw, 18 feet 6 inches.

The trial of the "Dakahlieh," the sister vessel, was made in Stokes' Bay:

The draft of water being, forward	15 feet 6 inches
Ditto aft ..	18 „
Displacement in Tons	2,200
Area of Midship section	484 square feet
Weights on Board	815 tons.

The power developed by these engines was upwards of 1,900 HP., with a three-bladed screw of 15 feet 9 inches diameter, and 19 feet 6 inches pitch, and with parallel blades—viz., of the same width at the boss as at the outer circle—gave the vessel a mean speed of 14·2 knots. This speed, is, we believe, unequalled for a screw-vessel of these proportions and dimensions, and with so large a cargo, and contrasts favourably, in point of results obtained by indicated power, with any of the vessels in the Peninsula and Oriental Company's fleet.

Plates 23 and 24.—These depict the plan and elevations of twin-screw engines, boilers, fittings, shafting, and propeller of 200 nominal horse-power collectively, constructed by Messrs. Watt, and fitted by them in the steam ships "Medusa" and "Triton." The engines are "direct acting," with the condenser between the shafts of both engines, by which location a compact arrangement results. The condenser is of the "injection" type, and internally arranged with care as to the correct disposition of the valves. The starting wheels are at the boiler side of the engines, thus enabling the engineer to proceed direct from the deck or stoking room to his post of duty without clambering over the engines or going around them. The boilers are the ordinary multitubular type, with a cylindrical superheater. The general arrangement of the piping, &c., are so evident from the drawings, that further description is scarcely requisite.

Plate 25.—This is a plan and elevation of an arrangement of a pair of "direct acting," by the above firm last alluded to, of the same collective nominal horse-power as the last example, fitted in H.M. steam ship "Research." The "condenser" is described and illustrated in page 261, the cylinders in page 276, and the expansion valve in page 301. The remainder of the arrangement and the main dimensions are obvious from the views depicted.

Plate 26.—This type of engine, by Messrs. Humphrys, is described in pages 52 to 56 inclusive; the "condenser" in page 248; the "air pump" valves in page 271; the "link motion" in page 290; and the "starting gear" in page 295.

Plate 27.—These engines are constructed by Messrs. Penn, and fitted in H.M.S. "Arethusa," being one of the *trial* ships, in which the makers of the engines were not bound as to terms or price as in the usual contracts. The "type" of "engine" is described in pages 42 and 43; the "cylinders" in page 274; the "slide valves" in

age 284; and the "condensers" in page 258, to which he firm has added the following:—

Nominal Horse-power	500
Indicated ditto on official trial, April 1st, 1862	2,871
Speed of Ship	12·694 knots
Diameter of Screw Propeller	18′
No. of Blades	2
Kind of Screw	Griffiths' patent
Pitch of ditto	(mean) 23′
Revolutions per minute	70
Pressure on Safety Valves,	20 lbs. per sq. inch
Vacuum	27½ to 28
Diameter of Cylinder	86½″
Effective diameter of ditto	80″
Diameter of Trunk	33″
Length of Stroke	3′ 6″
Diameter of Air Pumps	20″
Length of Stroke	3′ 6″
Diameter of Circulating Pumps	20″
Length of Stroke	3′ 6″
Diameter of Feed Pumps	5″
Length of Stroke	3′ 6″
Diameter of Condenser Tubes (outside)	1″
Length of ditto	6′ 9″
No. of ditto	4,832
No. of Boilers	4
Diameter of Tubes in ditto	3″
Length of ditto	7′
No.	1,216
No. of Furnaces	16
Breadth of ditto	2′ 10″
Length of ditto	7′
Diameter of Chimney	6′ 8″

Plate 28.—This is the most complete illustration in the present work, being an arrangement of a pair of return-action engines, by Messrs. Napier and Sons, comprising *six* views. As the illustrations are so complete it will be sufficient to state that the condenser is described in page 262, and that the following particulars added by the firm renders conclusion evident:—

Nominal Horse-power collectively	300
Diameter of Cylinder	63″
Length of Stroke	2′ 6″
Diameter of Piston-rods	5⅞″
Depth of Piston	8″
Diameter of Crank Shaft	12⅜″
Length of each Bearing of ditto ..	1′ 10″
Diameter of Connecting Rod (crank end)	7½″
Length of ditto	5′ 6″
" Guide Block Surface	2′ 6″
Width of ditto	16″
Diameter of Air Pump	16″
" Circulating Pump	16″
" Injection Pipe (1)	5½″
" Feed (1 to each engine) ..	3⅞″
" Snifting Valve	2½″
" Bilge Injection Cocks (2) ..	2½″
" Cylinder Relief Valves ..	5⅜″
" Blow-through ditto 10½ sq. ins. area to each engine.	
" Main Discharge Pipe	16″
" Exhaust Steam do. (each cylr.)	20″
" Single Supply do. "	15″
" Main ditto Pipe "	16″
Thickness of Cylinders "	1¼″
" Condensers "	1½″

Size of Suction Valves for Air Pump, rectangular 28 in. × 13 in.

No. of Valves (Suction) for Air Pumps, 1 to each end of Pump.

Discharge of Valves, 1 to each end of Pump.

No. of Suction Valves for Circulating Pump, 1 to each end of Pump.

Discharge of Valves, 1 to each end of Pump.

Suction and delivery valves are alike, and are the same for both air and circulating pumps. Each valve consists of two rectangular grated openings of 2 feet 3½ inches long by 5 inches wide.

Diameter of Condensing Tubes, ⅞ inch outside.

Tubes are about $\frac{1}{20}$th of an inch thick.

Length between tube-plates	6′ 3″
Diameter of Circulating Water-pipe ..	16″
" Discharge ditto	16″

No. of Tubes, 3,923.

Material of Air Pump Piston, brass.

" Feed Plunger, brass.

Feed Pump, single acting.

Distance between centres of Cylinders ..	8′ 9″
Diameter of Screw Propeller	15′
Pitch (mean)	14′ 6″

Kind of Propeller, Griffiths'.

" Expansion Valve, Gridiron.

Grade of Expansion, maximum, 2 from commencement of stroke.
Grade of Expansion, minimum, 6 from commencement of stroke.
Kind of Boilers, tubular.
No. of ditto, 4.
Length of Tubes in each, 6 feet 6 inches, and total number 272.
Length of Fire-grate, 5 feet 6 inches, breadth 3 feet 4 inches.
No. of Fire-Grates, 3.
Size of Funnel, 7 feet 9 inches by 3 feet 9 inches.

Plate 29.—Steam Launch engines and boiler, by Messrs. Rennie. The engines are vertical, direct acting, single cylinders, one to each screw shaft; surface condensers are attached to the sides of the boilers, the latter being the return tubular type. The following particulars, contributed by the firm, will be all that is necessary to appreciate the design in question:—

Kind of Boiler. Multitubular with return Tubes.	
Width of Shell of Boiler	2′ 5″
Length of ditto	4′ 9½″
No. of Tubes	40
Outer diameter of Tubes	1¾″
Length of ditto	3′ 6″
Width of Fire-grate	2′
Length of ditto	2′ 5½″
Diameter of Chimney	9″
„ Steam Dome	1′ 1″
No. of Steam Cylinders	2
Diameter of ditto	6″
Length of Stroke	6″
Diameter of Piston-rod	1″
„ Crank Shaft at Bearings ..	1¾″
Length of Bearings	2½″
Diameter of Feed Pumps	⅞″
Length of Stroke	6″
Diameter of Donkey Pump	1¼′
Length of Stroke	3″
No. of Tubes in Surface Condenser ..	86
Diameter of ditto ditto	$\frac{9}{16}$″
Mode of Circulating the Water, Centrifugal Pump.	

Screw Propellers, whether right or left-handed	Right and Left
Diameter of Screw	2′ 6″
Pitch of ditto	3′ 6″
Length of ditto	3¼″
No. of Blades	4
Length of Launch	42′
Breadth of ditto	10′ 11″

	Tons.	cwt.	qrs.	lbs.
Weight of Engines, Boiler, Centrifugal Pumps, Surface Condensers, Donkey Engine, &c., as lifted out	2	6	0	0
Stern Tube, Propellers, Shafts, Stern Brackets, Propeller Shafts, Thrust Bearings, Bolts, and Holding-down Plates		8	0	0
Sea Cocks and Discharge Pipes, Roses, Bolts, and Nuts		1	3	0
Coal Boxes, Spanner Racks, and Spanners		3	0	0
Floor Plates		2	0	0
	3	0	3	0
Water in Boilers		7	1	0
„ Condensers			3	0
	3	8	3	0
Engine Bearers of wood, and Chocks, forming part of the Boat		4	2	0
	3	13	1	0

It will be seen by the above that the total weight of the launch engine, including surface condensers, centrifugal pumps and their appurtenances, together with the water in the boiler and in the surface condensers, and of the wooden chocks and bearers fixed in the boat to carry the screw tubes, thrust block, boiler, and floor plates, is considerably under 4 tons, and excluding the surface condenser and its pipes, as in the launches hitherto adopted in the English navy, the weight will be 8 cwt. less, or 3 tons 5 cwt. 1 qr. altogether. This small additional weight of 8 cwt. is the substitute in these engines for the cumbrous tanks of fresh water placed in the ordinary launch, the weight of which, with the water in them is upwards of a ton.

Trial of H.M. Steam-Launch No. 10, fitted with High-Pressure Engines and Surface Condensers by Messrs J. and G. Rennie.

	Full Power, High Pressure, without Condensers.	Full Power, Condensing, using Condensers.
When tried	8th June, 1866	
Where tried	Stokes Bay.	
Nominal Horse Power	5 Horses.	
Makers' Name	J. & G. Rennie.	
Draught of Water, Forward	1′ 11″	
Draught of Water, Aft	3′ 0½″	
Load on Safety Valve	80 lbs.	
Pressure of Steam in Boilers	80·833 lbs.	73·25 lbs.
Number of Revolutions of Engines, Maximum	330	340
Number of Revolutions of Engines, Mean	326·333	328·5
Mean Pressure in Cylinders	54·958 lbs.	58·344
Indicated Horse Power	30·732	32·84
Speed of Vessel	7·897 knots.	8·054 knots.
Quantity of Coal on board	5 cwt.	
Quantity of Stores	1 day.	
Propeller, Description	Common 4-bladed and variable pitch.	
Propeller, Diameter	2′ 6″	
Propeller, Pitch	3′ 6″	
Propeller, Length	3¼″	
Propeller, Immersion of upper edge	3″	
True mean Speed of 6 runs with and against tide, in knots	7·897	
True mean Speed of 4 runs with and against tide, in knots	..	8·054

Plate 30.—This is an example of paddle-wheel engines by Messrs. Watt; the description of the details will be found in Chapter V., commencing in page 193.

Plate 31.—Messrs. Maudslay's example of boiler and twin screw engines for H.M. Launches is illustrated by this plate. The cylinders are secured direct to the fire-box shell, and the shafts are supported by brackets secured below the cylinders; the latter being also sustained by stays connected to the brackets. A donkey engine feed pump is connected to the smoke box, and the engine feed pumps are worked by cranks at the extremities of the shafts. The following particulars by the firm are sufficient to understand the practicability of the arrangement in question:—

Kind of Boiler—Locomotive.	
Diameter of ditto	2′ 1″
Length of ditto	3′ 7″
No. of Tubes	31
Outer diameter of Tubes	2″
Length of ditto	3′ 7″
Width of Fire-grate	2′ 1″
Length of ditto	2′ 3″
Diameter of Chimney	10″
„ Steam Dome	1′ 3½″
No of Steam Cylinders	4
Diameter of ditto	5″
Length of Stroke	6″
Width of Supply Port	7/16″
„ Exhaust Port	1½″
Length of ditto	2½″
Width of opening caused by Slide	¼″
Outside Lap	⅜″
Diameter of Piston Rod	1″
„ Crank Shaft at Bearings	2″
Length of Bearings	3½″
Diameter of Feed Pumps	1¾″
Length of Stroke	2½″
Diameter of Donkey Pump	1⅜″
Screw Propeller, 1 right handed and 1 left.	
Diameter of Screw	2′ 6″
Pitch of ditto	3′
Length of ditto	3″
No. of Blades	4
„ Revolutions	306 to 375
Speed of Launch, 7·25 knots per hour.	
Steam pressure	65 lbs.
Length of Launch	42′
Breadth of Ditto	10′ 6″
Depth of Ditto	3′ 9″

	Tons.	cwts.	qrs.
Weight of Engines and Boiler complete	1	19	2
Weight of Water in Boiler	0	7	2
Total	2	7	0

Plate 32.—These are elevations and plan of a boiler and twin-screw engines for one of H.M. Launches, by Messrs. J. Penn and Son. The boiler is the locomotive type; the engines are secured to the sides at the fire-box end, and thus frames are dispensed with. The donkey pump is attached to the smoke box; the form of the engines, pumps, and shafting is obvious from the illustrations, without further notice, excepting the following, by the firm:—

Kind of Boiler, Locomotive.

Number of Tubes	35
Outside Diameter	2¼′
Length, ditto	3′ 10″
Width of Fire-grate	2′ 4″
Length	2′
No. of Steam Cylinders	4
Diameter of "	5″
Length of Stroke	6″

Screw Propellers (2), Right and Left-handed.

Diameter	2′ 6″
Pitch	4′
No. of Blades in each	4
Length of Boat	42″

Ship's Launch.

Total weight of Machinery, Boiler, Water and Wood chocks, and Engine bearers, complete, 3 tons 4 cwt. 2 qrs.

Indicated Power, 36 to 40 (collectively).

Revolutions, 300.

Speed, 8 to 8½ knots.

Plate 33.—The engines for H.M.S. "Lord Clyde" are illustrated by this plate, being the contribution of Messrs. Ravenhill and Hodgson. The type of engines is "return action," the cylinders being 116 inches in diameter, and the stroke of piston 4 feet; the cylinders are jacketed, and the grade of expansion is determined by the ordinary "gridiron valve," as adopted by Messrs. Watt and Penn. The condensers are the "surface" system, the water being circulated by centrifugal pumps, driven by auxiliary engines. Steam starting gear is introduced to raise the link; and thus also starts the engines when the "hand" power is futile.

Plate 34.—This is a pair of twin-screw engines, and locomotive boiler, for the steam Launch lately supplied by Messrs. Forrester and Co., Liverpool, for the S.S. "Great Eastern;" the following description by the firm will readily convey a correct idea of the arrangement:—

The boiler is of the locomotive type; the barrel is 2 feet 6 inches diameter, and 5 feet long, including smoke box. There are 51 tubes, of 1¾ inches outside diameter, and 4 feet long, giving a heating surface of 93 square feet; the outer fire-box is 2 feet 6 inches long, by 2 feet 11½ inches wide; the area of the fire-grate is 5 square feet. The diameter of the chimney is 9 inches, and its height, from top of smoke-box, is 7 feet. The dome is 16 inches diameter, and 2 feet high, and contains a trap to prevent water from entering the steam-pipe when the boiler primes. The safety-valves are two in number, are loaded by spring-balances to 60 lbs. pressure per square inch. The steam cylinders, of which there are two to each screw shaft, are 5 inches diameter, and 6 inches stroke of piston. The slide-valves are single ported, having a lap of $\frac{5}{16}$ inch, and open for steam ¼ inch; the ports are $\frac{7}{16}$ inch wide, so that a free exhaust is permitted. The forked connecting rods are 15½ inches long; and the piston rods, each 1 inch in diameter, are guided by brackets below the crosshead pin. The crank shaft is supported by the capped ends of wrought-iron columns, the upper ends of which are fixed to the cylinders, and the lower ends bolted to the fire-box. The engines are fixed to wrought-iron brackets, which are planed true after being rivetted to the boiler. The feed pumps, one to each engine, are worked by eccentrics, and are 1⅜ inches diameter, and 2¾ inches stroke. All eccentric rods are of gun-metal, cast in one piece with the strap; the eccentric pulleys for the valve motion are forged with the shaft, and turned out of the solid. The donkey pump is bolted to the side of the smoke-box, its steam cylinder is 3 inches diameter; the pump plunger is 1½ inch diameter, by 3 inches stroke; the donkey is also adapted for being worked by hand.

The screw-propellers are four-bladed, each being 3 feet diameter and 3½ feet pitch. The propellers, propeller shafts, and stern tubes are of gun-metal.

Gun-metal bosses are rivetted in with the plates of the boat, for supporting the inner end of the stern tubes,

the outer ends of which are fixed in wrought-iron brackets close to the propellers, so that the shafts are entirely protected outside the boat.

The launch is 50 feet long, and 10 feet beam, and built entirely of steel plates, and also steel frames.

Plate 35.—Marine boilers have not been varied so much as the engines, simply because the requirements are more easily attained in the former than the latter. The plate under notice illustrates an example by Messrs. Dudgeon, fitted by them in the steam ship "Ruahine." To enable the "staying" portions to be clearly understood, they have been represented by *blue* lines. The arrangement of the fire boxes, tubes, and uptake is the ordinary kind, and the "superheater" is fully described in pages 157 and 159. The only features in the design presenting any novelty worthy of comment are the safety valve weights *above* the casing and the mode of lifting the valves, the remainder of the arrangement being the general practice: as the drawing is freely dimensioned the proportions can be fully understood.

Plate 36.—This is the complete plan of the engines fitted by Messrs. Penn in H.M. steam ship "Achilles," 1250 nominal horse-power collectively. This arrangement is explained in detail in pages 43 and 44.

SYNOPTICAL INDEX OF PLATES.

Plate 1.—Three cylinder expansive engines, 300 H.P. collectively, fitted in the Imperial Russian iron-cased Frigate "Pervenelz," by Messrs. Maudslay, Sons and Field.

Plate 2.—Three cylinder expansive engines, 300 H.P. collectively, fitted in the Imperial Russian iron-cased Frigate "Pervenelz," by Messrs. Maudslay, Sons, and Field.

Plate 3.—Supplementary inverted engine and surface condenser, by J. F. Spencer, Patentee, for I.S.S. "Frankfort."

Plate 4.—Supplementary inverted engine and surface condenser, by J. F. Spencer, Patentee, for I.S.S. "Frankfort."

Plates 5 & 6.—Details of Supplementary inverted engine and surface condenser, by J. F. Spencer, Patentee for I.S.S. "Frankfort."

Plate 7.—Combined pump for supplementary inverted engine and surface condenser, by J. F. Spencer, Patentee, for I.S.S. "Frankfort."

Plate 8.—Details of supplementary inverted engine and surface condenser, by J. F. Spencer, Patentee, for I.S.S. "Frankfort."

Plate 9.—Twin screw Engines, of 350 H.P. collectively, fitted in the M.S. "Ruahine," constructed by Messrs. J. and W. Dudgeon.

Plate 10.—End elevation of the engines fitted in the M.S. "Ruahine," constructed by Messrs. J. and W. Dudgeon.

Plate 11.—Pistons of the engines fitted in the M.S. "Ruahine," constructed by Messrs. J. and W. Dudgeon.

Plate 12.—Cylinder of the engines fitted in the M.S. "Ruahine," constructed by Messrs. J. and W. Dudgeon.

Plate 13.—Slide valve and casing of the engines fitted in the M.S. "Ruahine," constructed by Messrs. J. and W. Dudgeon.

Plate 14.—Condensers of the engines fitted in the M.S. "Ruahine," constructed by Messrs. J. and W. Dudgeon.

Plate 15.—Transverse section of H.M. armour-cased iron ship "Hector," showing engines, 800 H.P. nominal, fitted by Messrs. R. Napier and Sons, Glasgow.

Plate 16.—Air pump trunk engine, 800 H.P. collectively, fitted in H.M. armour-cased ship "Hector," constructed by Messrs. R. Napier and Sons, Glasgow.

Plate 17.—Elevation of cylinders of the engines fitted in H.M. armour-cased ship "Hector," constructed by Messrs. R. Napier and Sons, Glasgow.

Plate 18.—Elevations of condensers of the engines fitted in H.M. armour-cased ship "Hector," constructed by Messrs. R. Napier and Sons, Glasgow.

Plate 19.—Return connecting rod, double piston rod, engines, 900 H.P. collectively, fitted in the S.S. "Roma and Venetia," constructed by Messrs. Maudslay, Sons and Field.

Plate 20.—Return connecting rod, double piston rod, engines, 900 H.P. collectively, fitted in the S.S. "Roma and Venetia," by Messrs. Maudslay, Sons and Field.

Plate 21.—Return connecting rod, double piston rod, engines, 900 H.P. collectively, fitted in the S.S. "Roma and Venetia," by Messrs. Maudslay. Sons and Field.

Plate 22.—Double piston rod, engines and boilers, 350 H.P. collectively, expansive surface condensing and superheating. fitted on board the Egyptian Government steamers "Charkieh" and "Dakahlieh," constructed by Messrs. J. and G. Rennie, London.

Plate 23.—Engines and boilers, 200 H.P. collectively,

fitted in the twin screw armour-plated S.S. "Medusa and Triton," by Messrs. James Watt and Co.

Plate 24.—Transverse sectional elevations of twin screw armour-plated S.S. "Medusa and Triton," constructed by Messrs. James Watt and Co.

Plate 25.—Direct acting engines, 200 H.P. collectively, fitted in H.M.S. "Research," constructed by Messrs. James Watt and Co.

Plate 26.—Direct acting engines, 400 H.P. collectively, constructed by Messrs. Humphrys and Tennant.

Plate 27.—Expansive engines, of 500 H.P., fitted in H.M.S. "Arethusa," constructed by Messrs. John Penn and Son.

Plate 28.—Expansive engines, 300 H.P., with surface condensers, constructed by Messrs. R. Napier and Sons.

Plate 29.—Twin screw engines, boiler, and surface condensers, of one of H.M.S. Launches, by Messrs. J. and G. Rennie.

Plate 30.—Boilers and oscillating paddle-wheel engines, 300 H.P. collectively, fitted in the I.S.S. "Abigail,' constructed by Messrs. James Watt and Co.

Plate 31.—Twin screw engines and boiler of one of H.M.S. Launches, by Messrs. Maudslay, Sons, and Field.

Plate 32.—Twin screw engines and boiler of one of H.M.S. Launches, by Messrs. J. Penn and Son.

Plate 33.—Arrangement of engines of 1000 H.P. for H.M.S. "Lord Clyde," by Messrs. Ravenhill, Hodgson, and Co.

Plate 34.—Twin screw engines and boiler for the Steam Launch of the "Great Eastern," by Messrs. Forrester and Co.

Plate 35.—Boiler by Messrs. Dudgeon, fitted in S.S. "Ruahine."

Plate 36.—Plan of engines, by Messrs. Penn, fitted in H.M.S. "Achilles."

SYNOPTICAL INDEX OF SUBJECTS.

ARRANGEMENT OF ENGINES.

OSCILLATING PADDLE-WHEEL ENGINES.

SCREW PROPELLER ENGINES.

BOILERS.

SUPERHEATERS.

SALINOMETERS.

DETAILS OF OSCILLATING ENGINES.

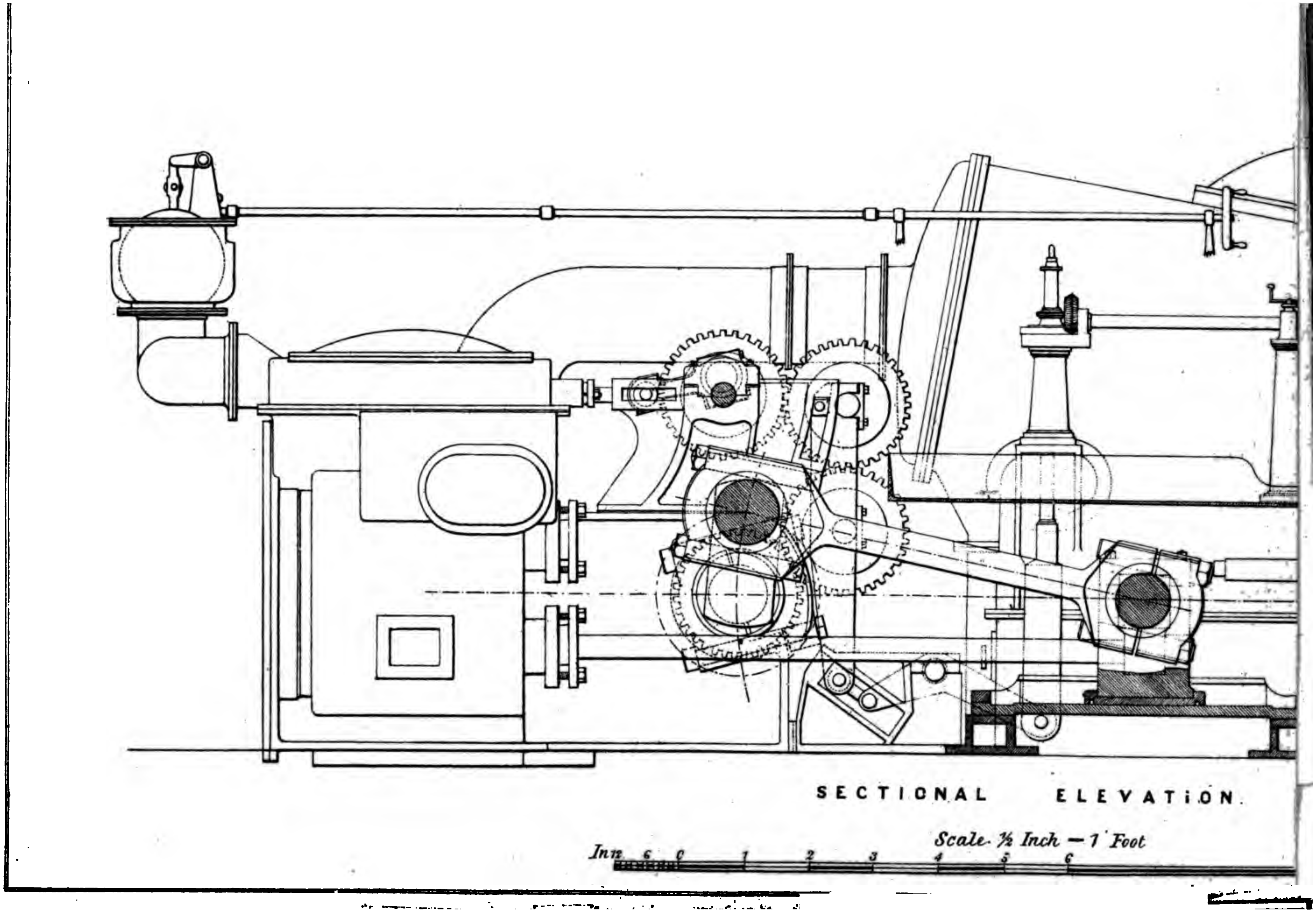
DIAGRAMS OF THE PRINCIPLE.

INDICATOR DIAGRAMS.

INDEX OF SUBJECTS.

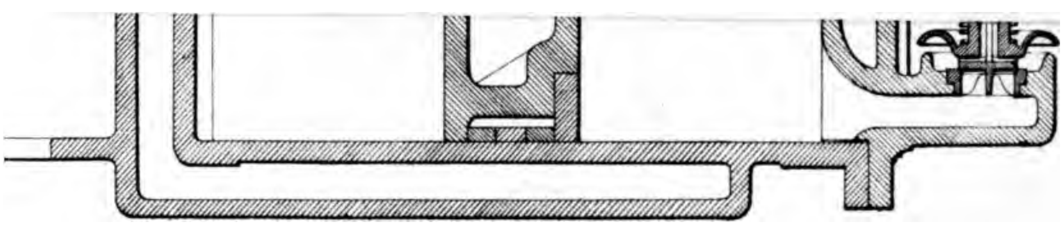
INDEX OF ILLUSTRATIONS.

LONDON:
PRINTED BY WILLIAM CLOWES AND SONS,
STAMFORD STREET AND CHARING CROSS.



SECTIONAL ELEVATION.

Scale ½ Inch — 1 Foot



Feet

PLATE 3

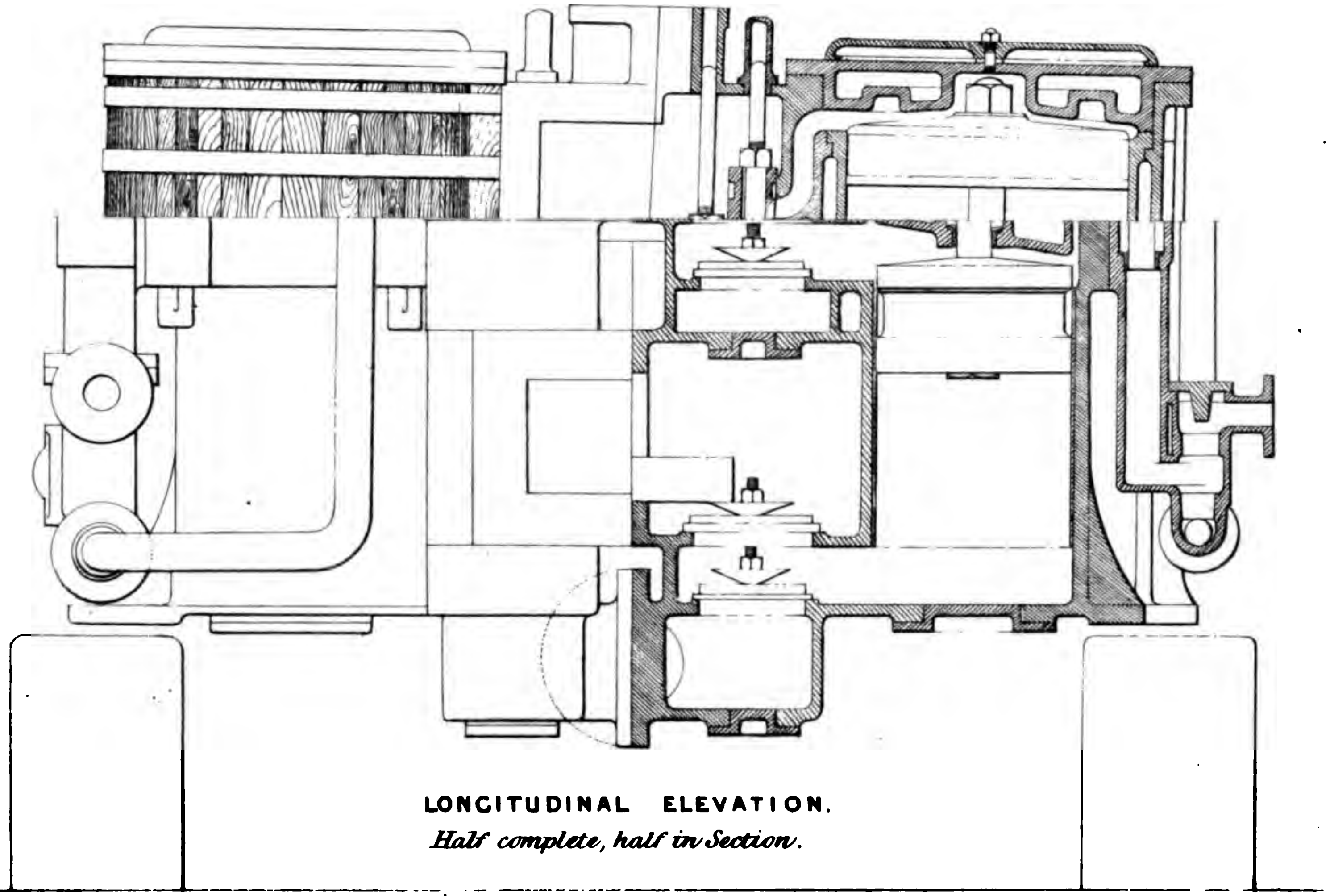
SUPPLEMENTARY INVERTED ENGINE AND SURFACE CONDENSER,

by

J. F. SPENCER, PATENTEE,

for

I.S.S. FRANKFORT.



LONGITUDINAL ELEVATION.

Half complete, half in Section.

N. P. Burgh Direx

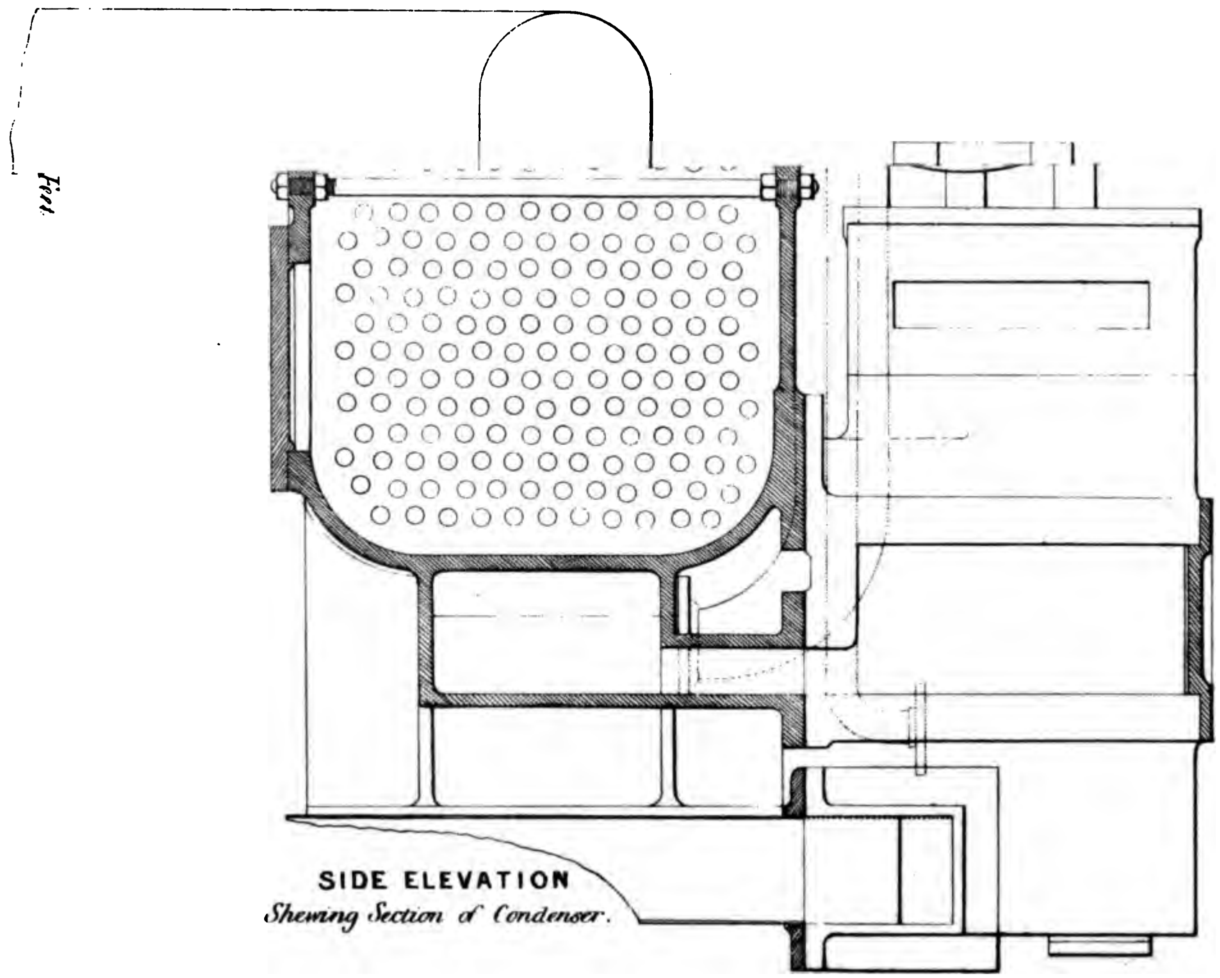
SUPPLEMENTARY INVERTED ENGINE AND SURFACE CONDENSER.

by

J. F. SPENCER, PATENTEE,

for

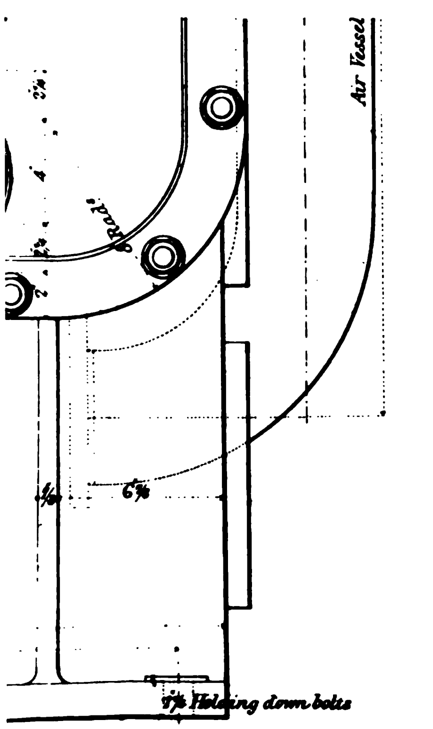
"I.S.S. FRANKFORT".



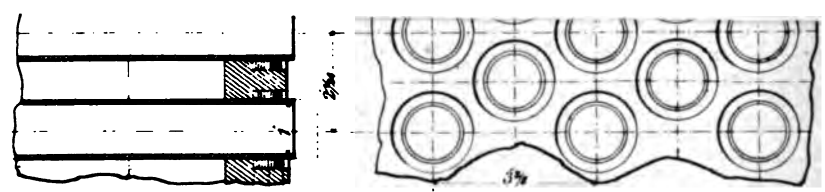
SIDE ELEVATION

Shewing Section of Condenser.

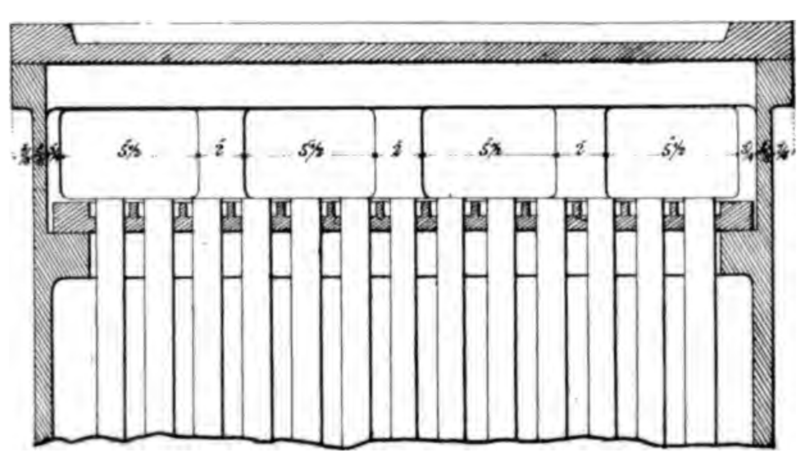
N. P. Burgh Direx



SER.

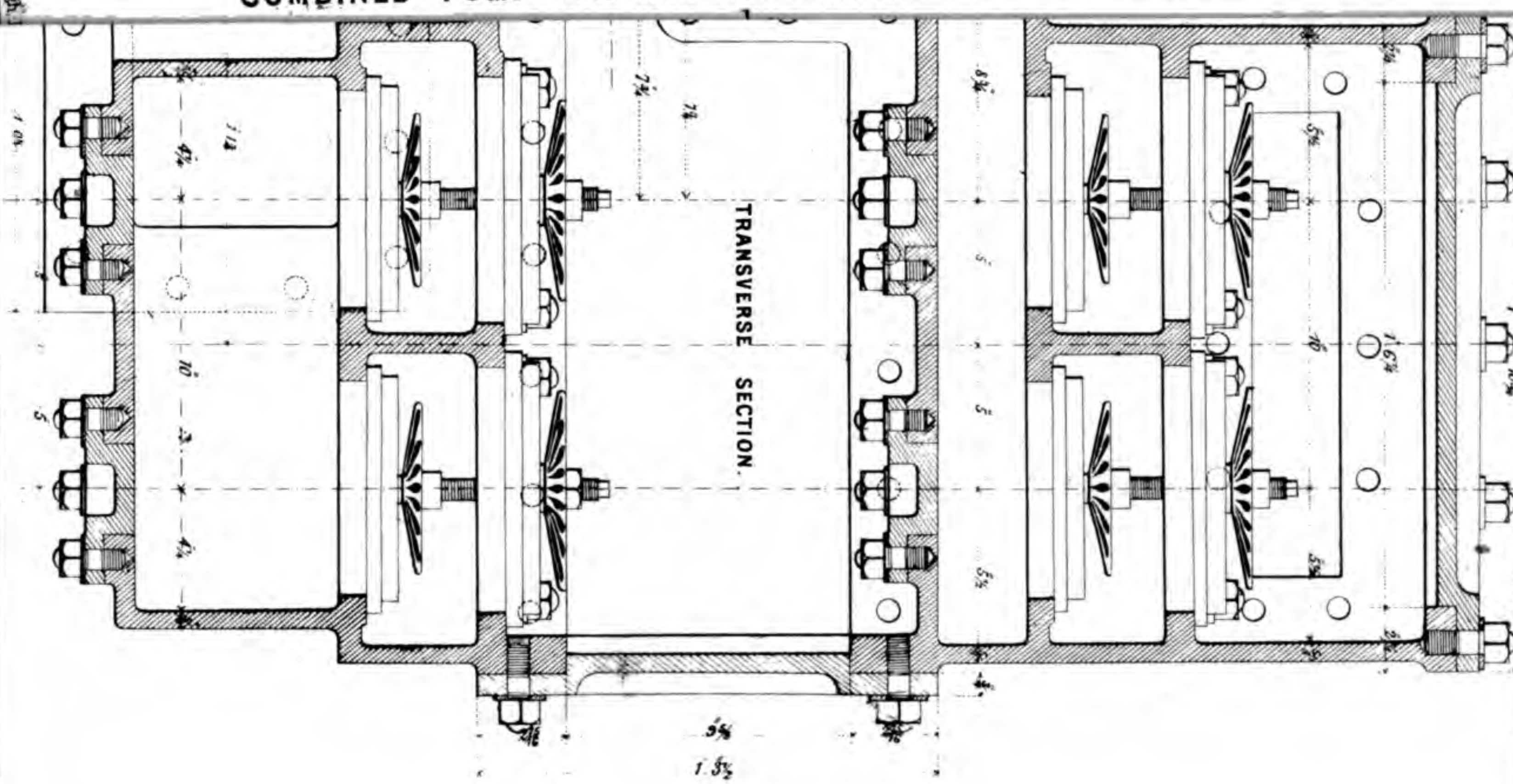


SECTION SHEWING TUBE PACKING.
Quarter Size.



SECTIONAL PLAN AT A.

COMBINED PUMP FOR SUPPLEMENTARY INVERTED ENGINES.



Scale 2 Inches – 1 Foot.

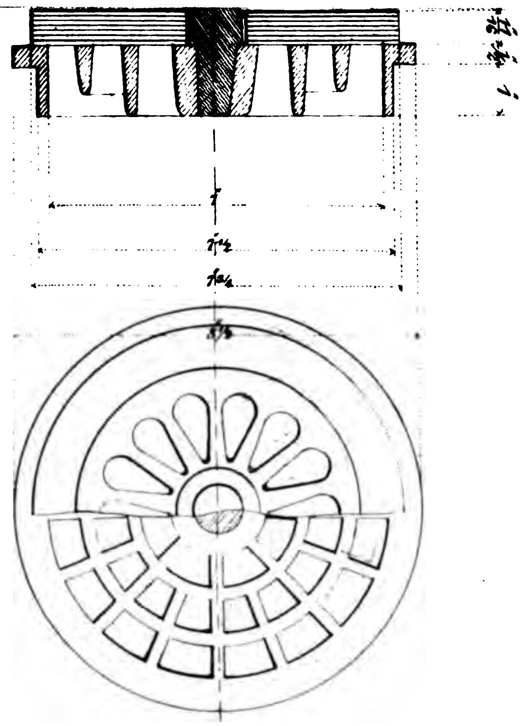
N. P. Burgh.

DETAILS OF COMBINED PUMP FOR SUPPLEMENTARY INVERTED ENGINES.

by

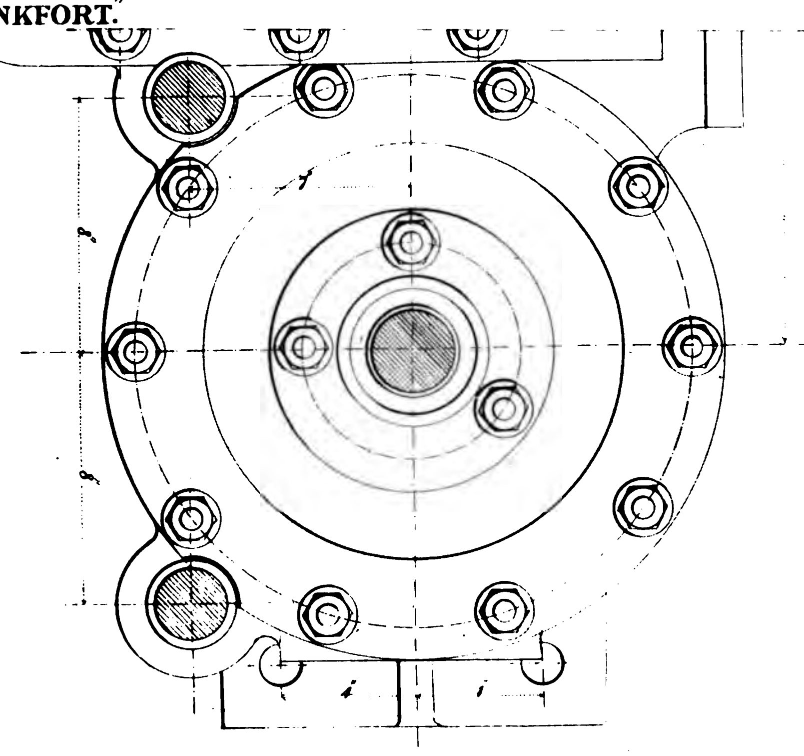
J.F. SPENCER, PATENTEE.

for "I.S.S. "FRANKFORT."



SUCTION AND DISCHARGE VALVES.

Scale. 3 Inches – 1 Foot.



Half Complete

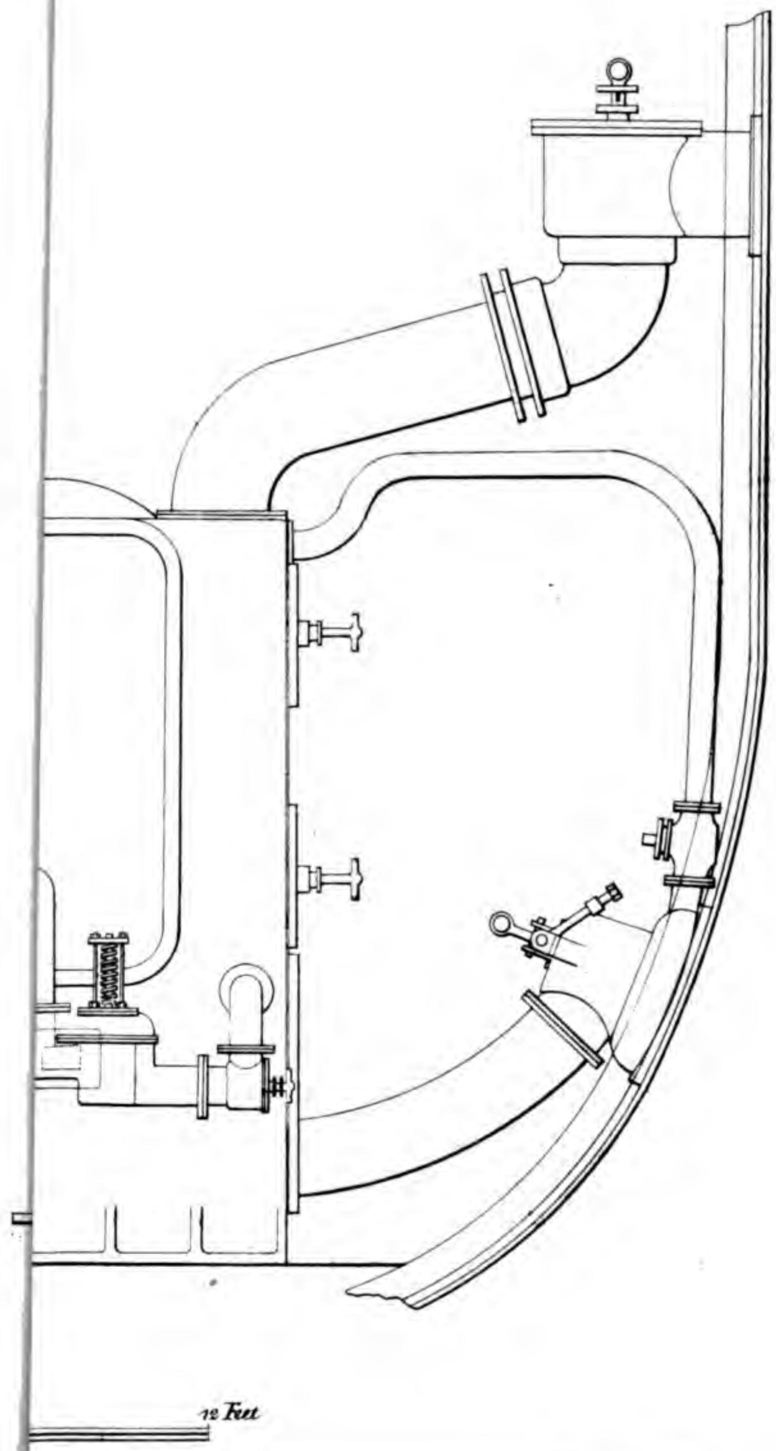
PLAN OF PUMPS.

N. P. Burgh. Direx.

-ale, ½ Inch = 1 Foot.

6 9 12 Feet

HINE."

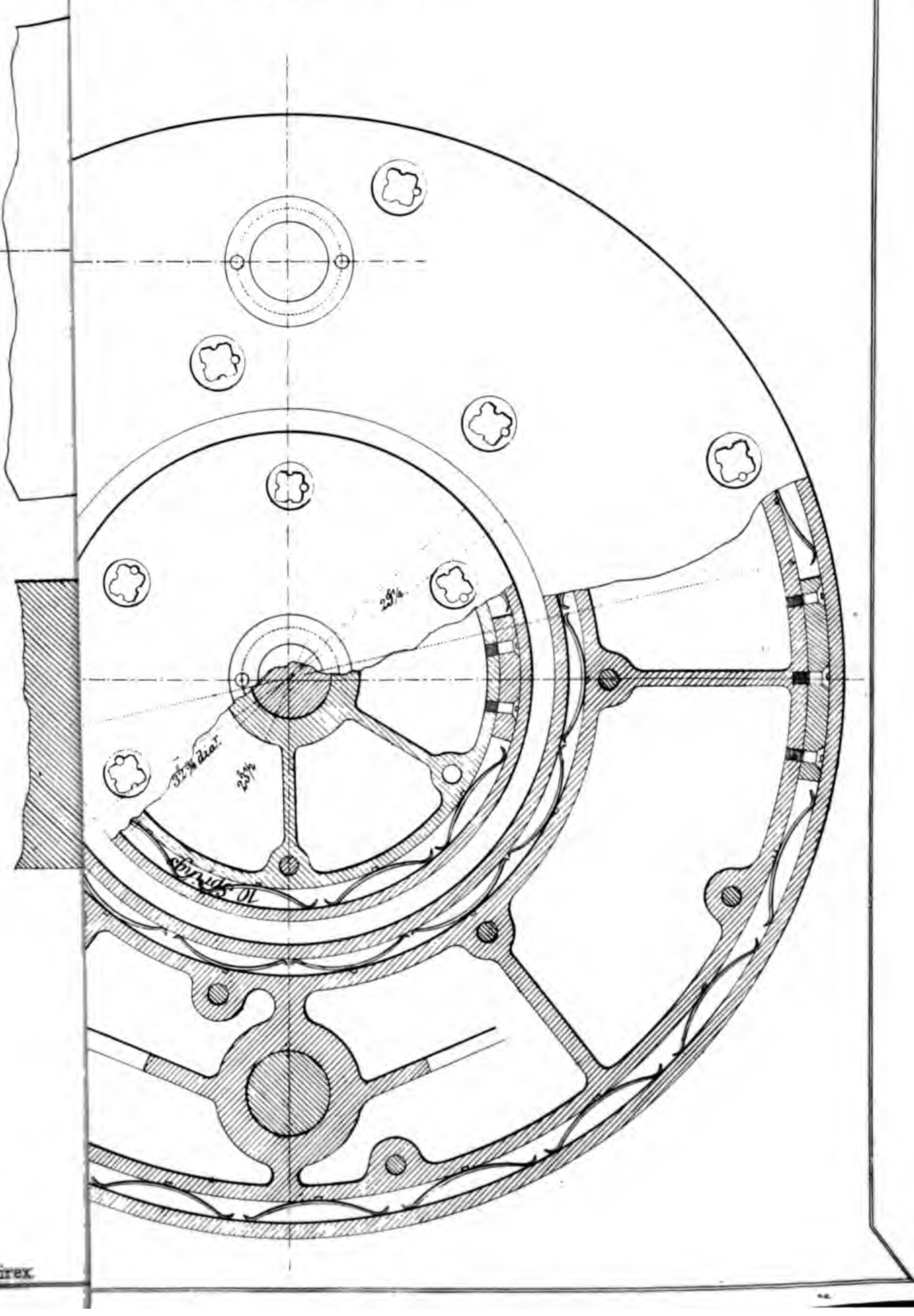


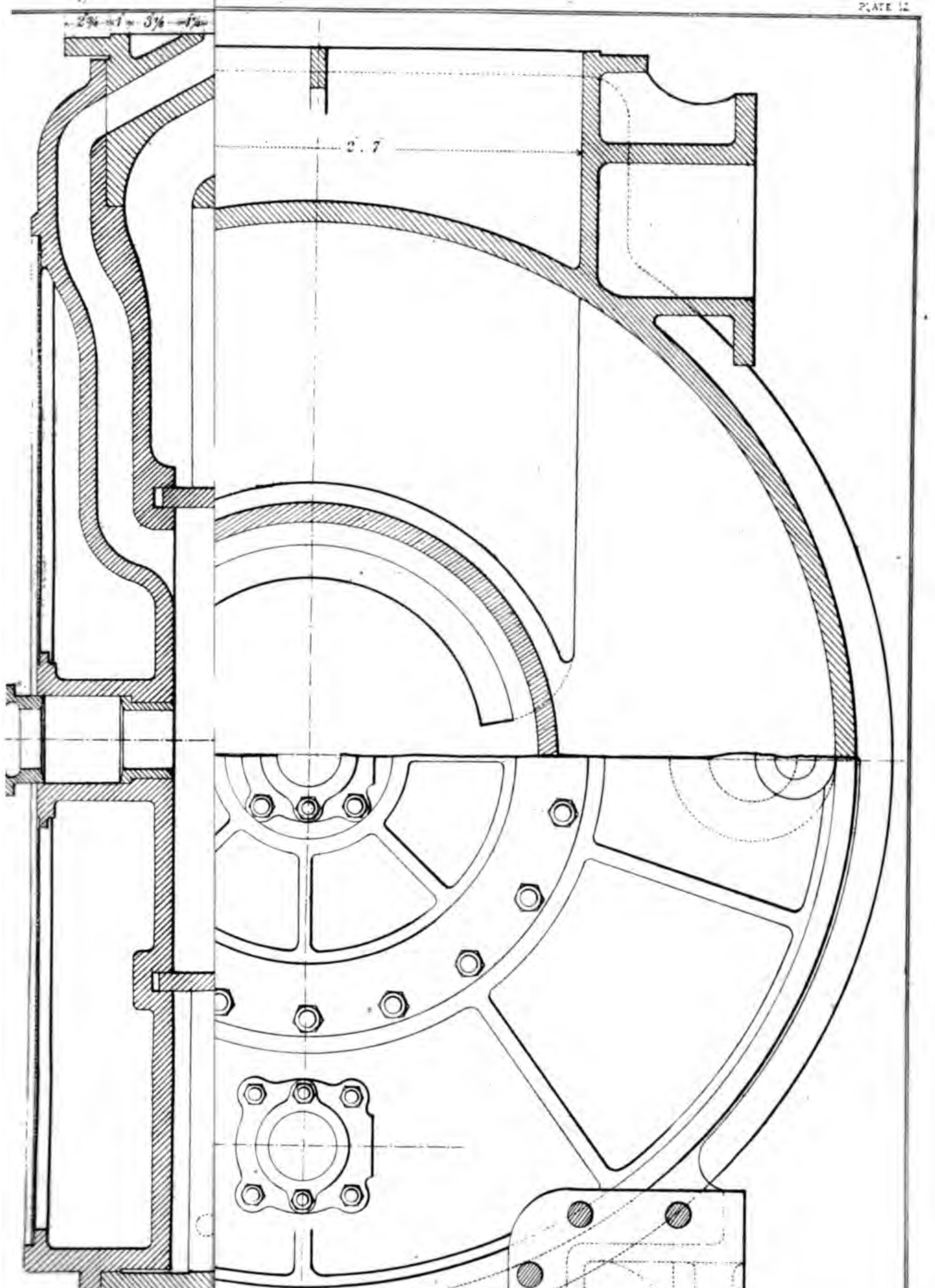
N.P. Burgh Del

MESSRS J & W. DUDGEON.

Mode of Securing "Face Ring"

Full Size.





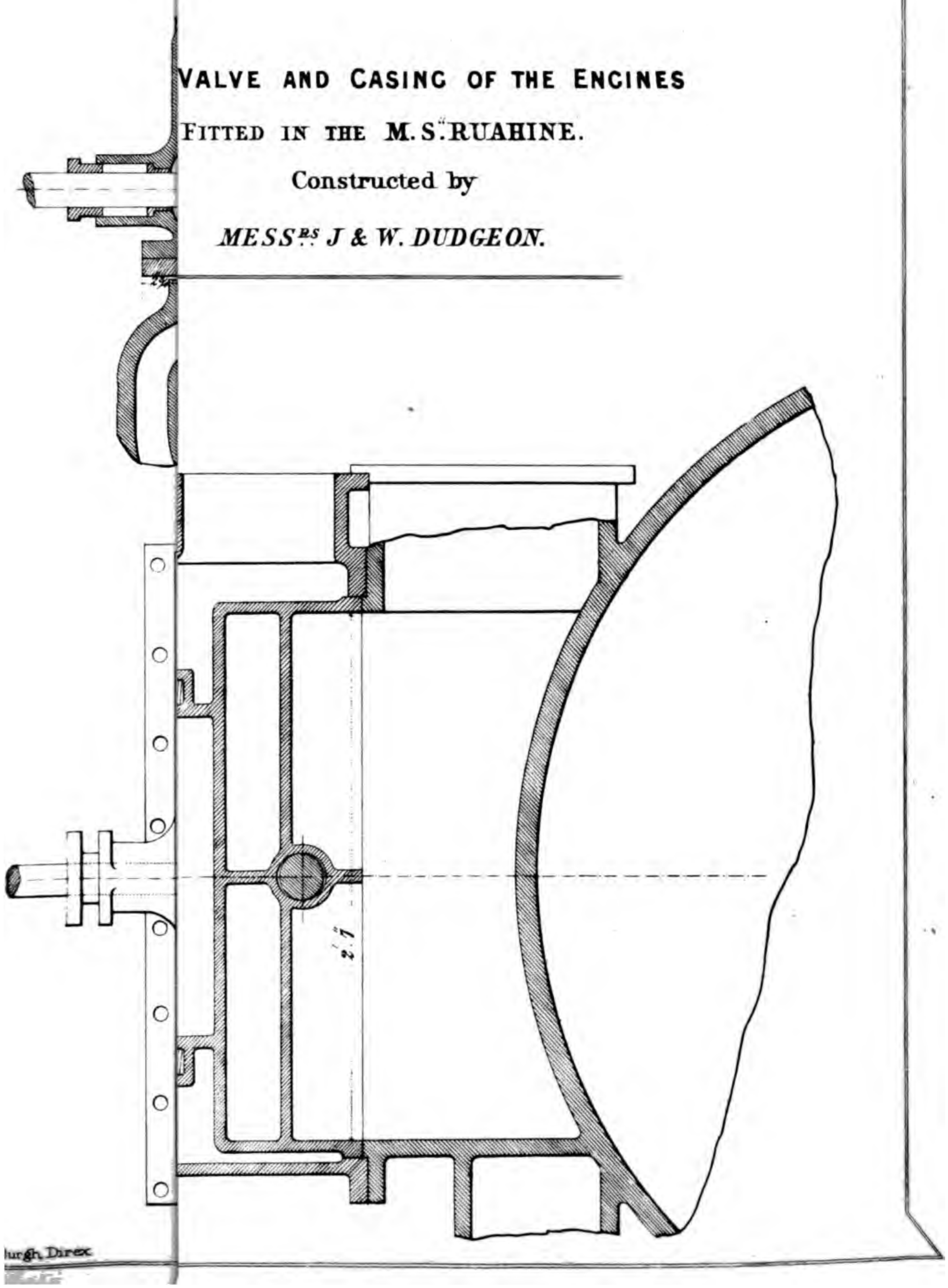
Burgh, Direx

VALVE AND CASING OF THE ENGINES

FITTED IN THE M. S.ˢ RUAHINE.

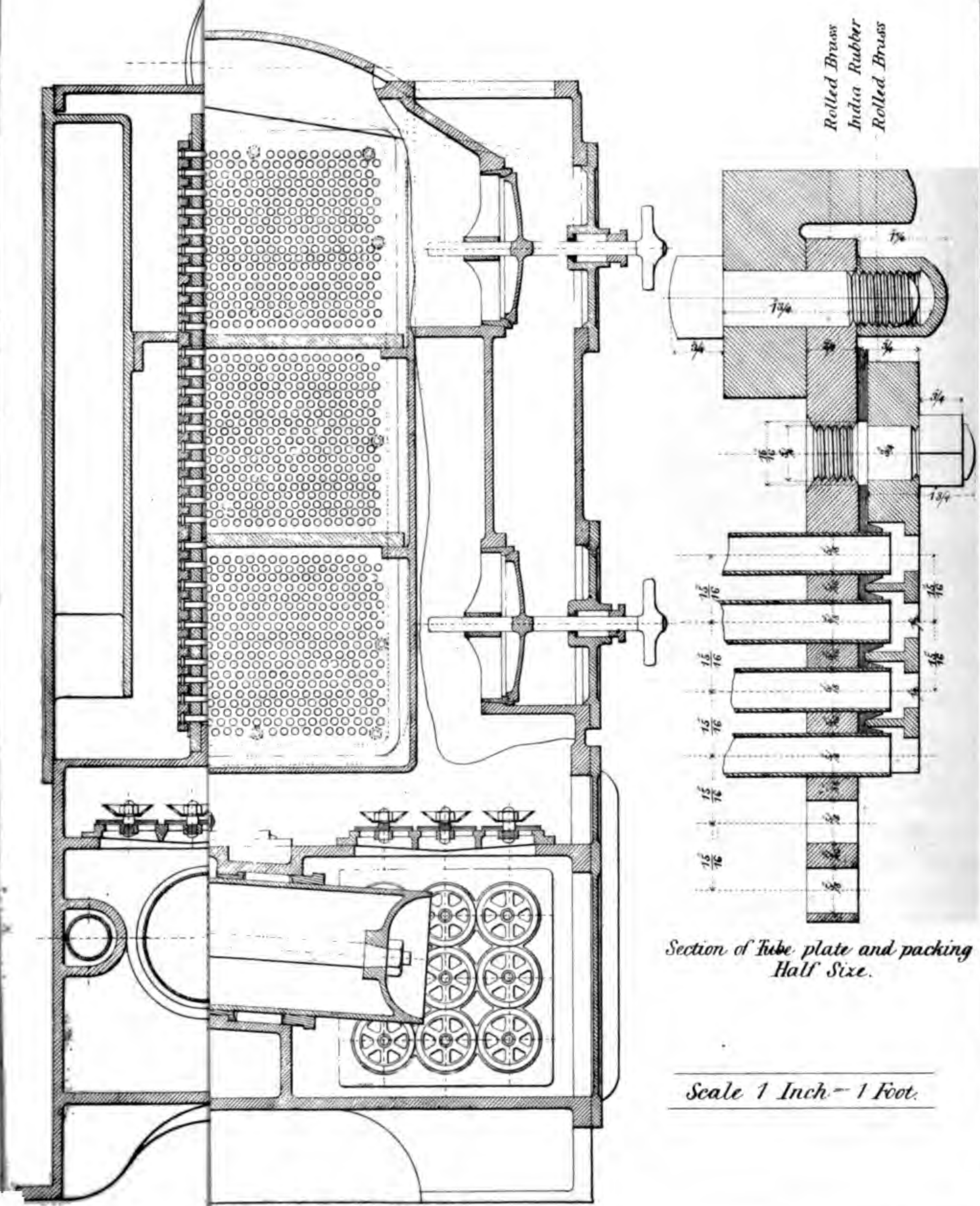
Constructed by

MESS.RS J & W. DUDGEON.



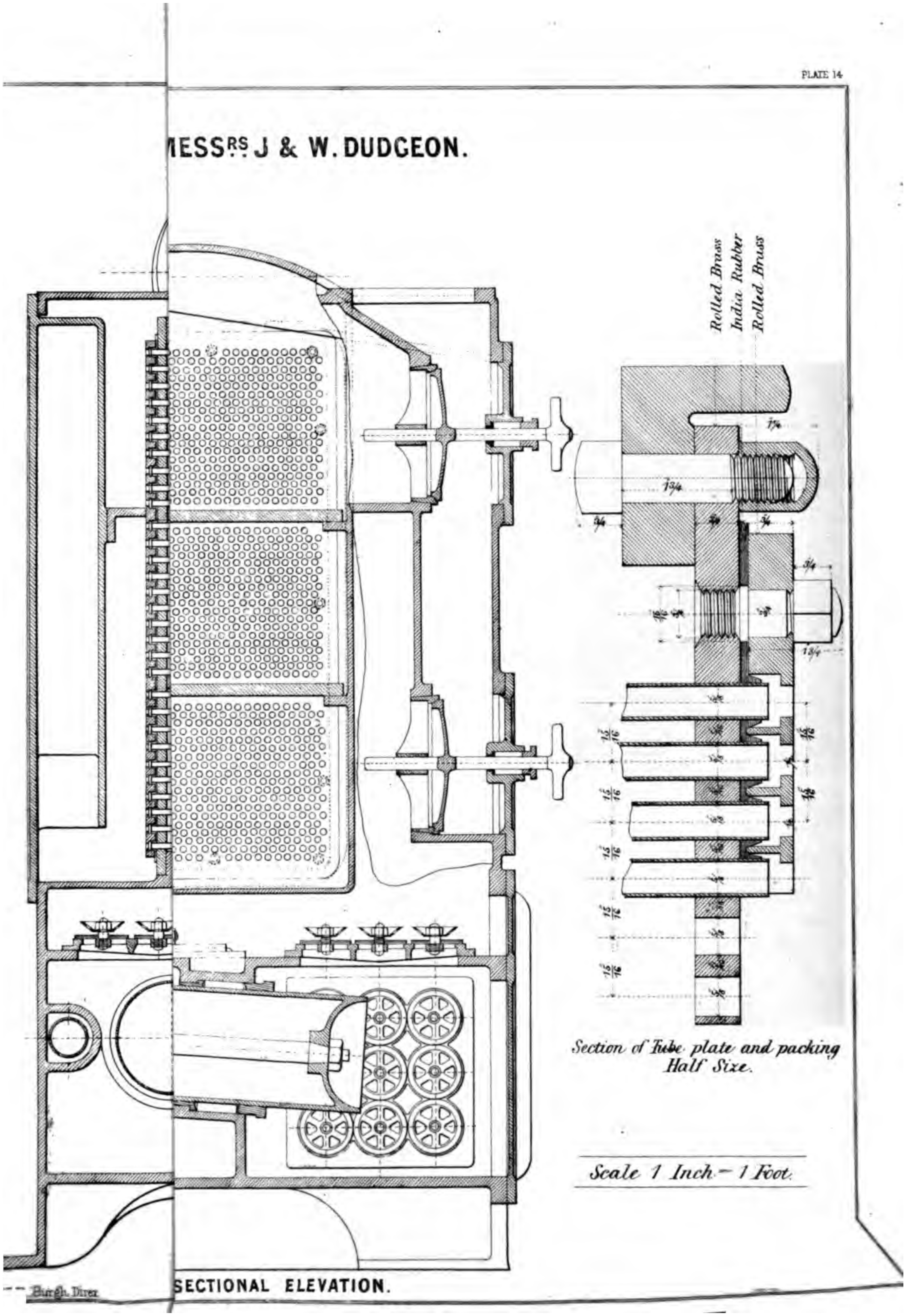
urgh, Direx

MESSRS J & W. DUDGEON.



Dirox

SECTIONAL ELEVATION.


MESSRS. J & W. DUDGEON.
Rolled Brass
India Rubber
Rolled Brass
Section of Tube plate and packing
Half Size.
Scale 1 Inch = 1 Foot.
SECTIONAL ELEVATION.
Burgh. Direx.

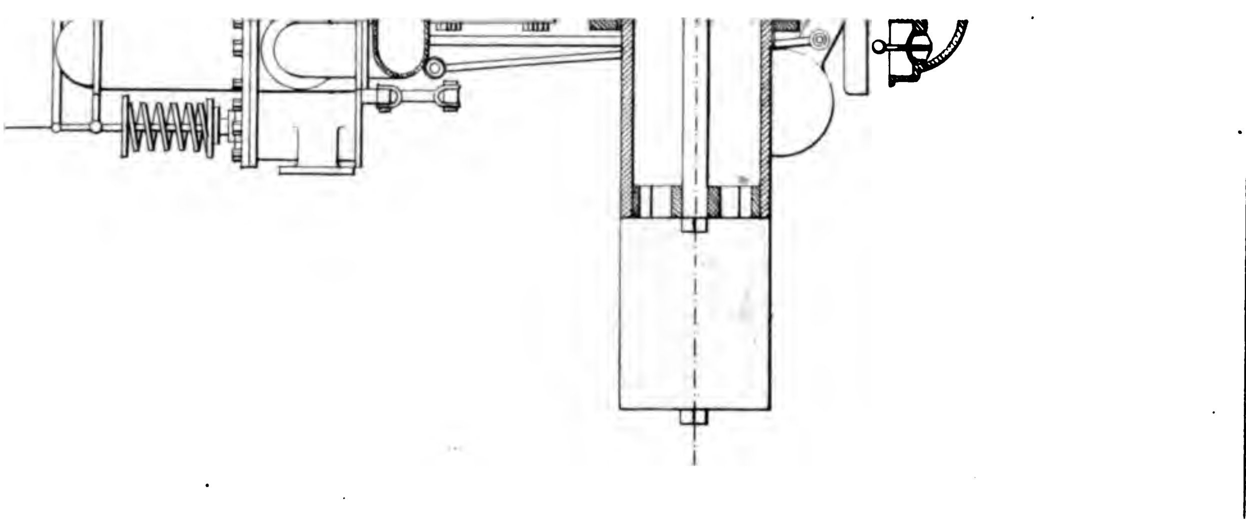
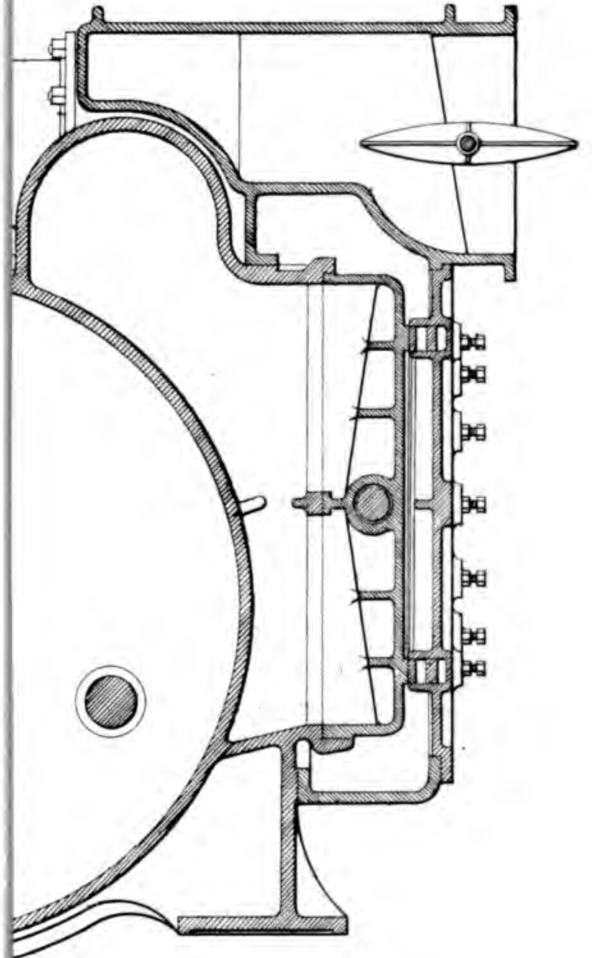


PLATE 17.

.S. HECTOR".



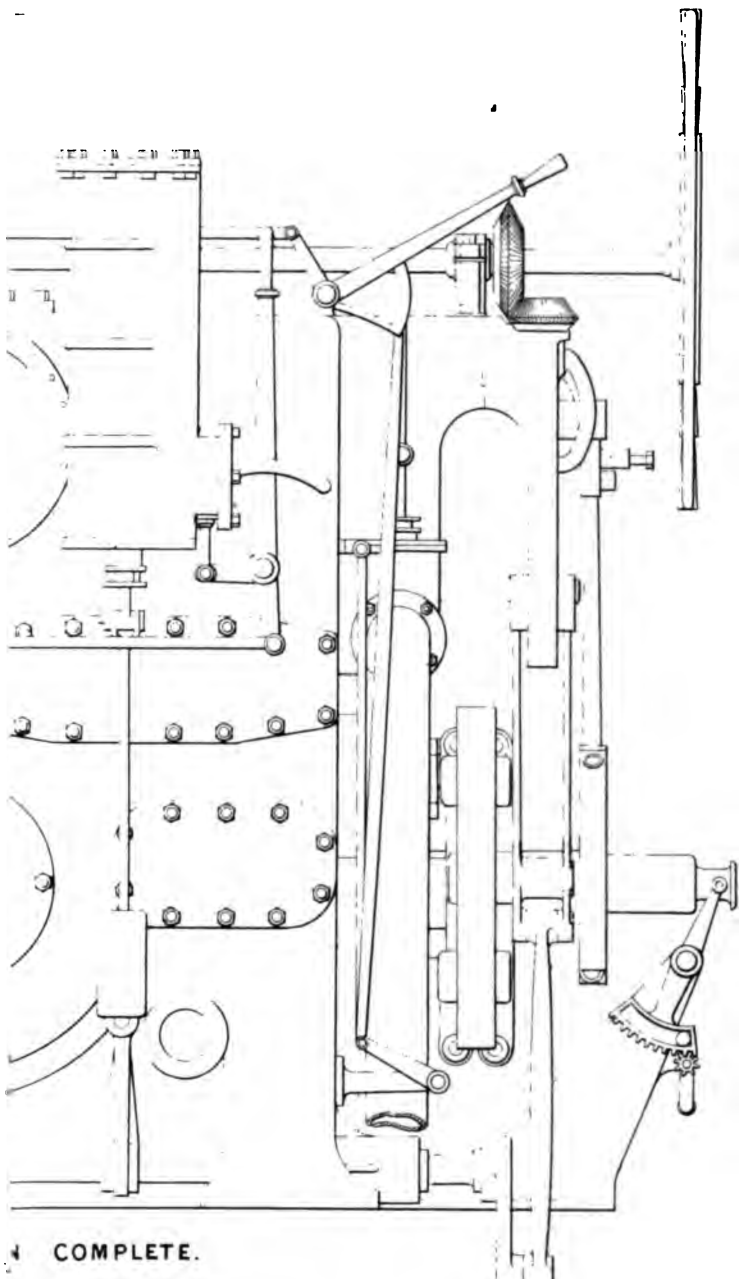
VERSE SECTIONAL ELEVATION.

12 Feet

N.P. Burgh, Dir

S.

GLASGOW.



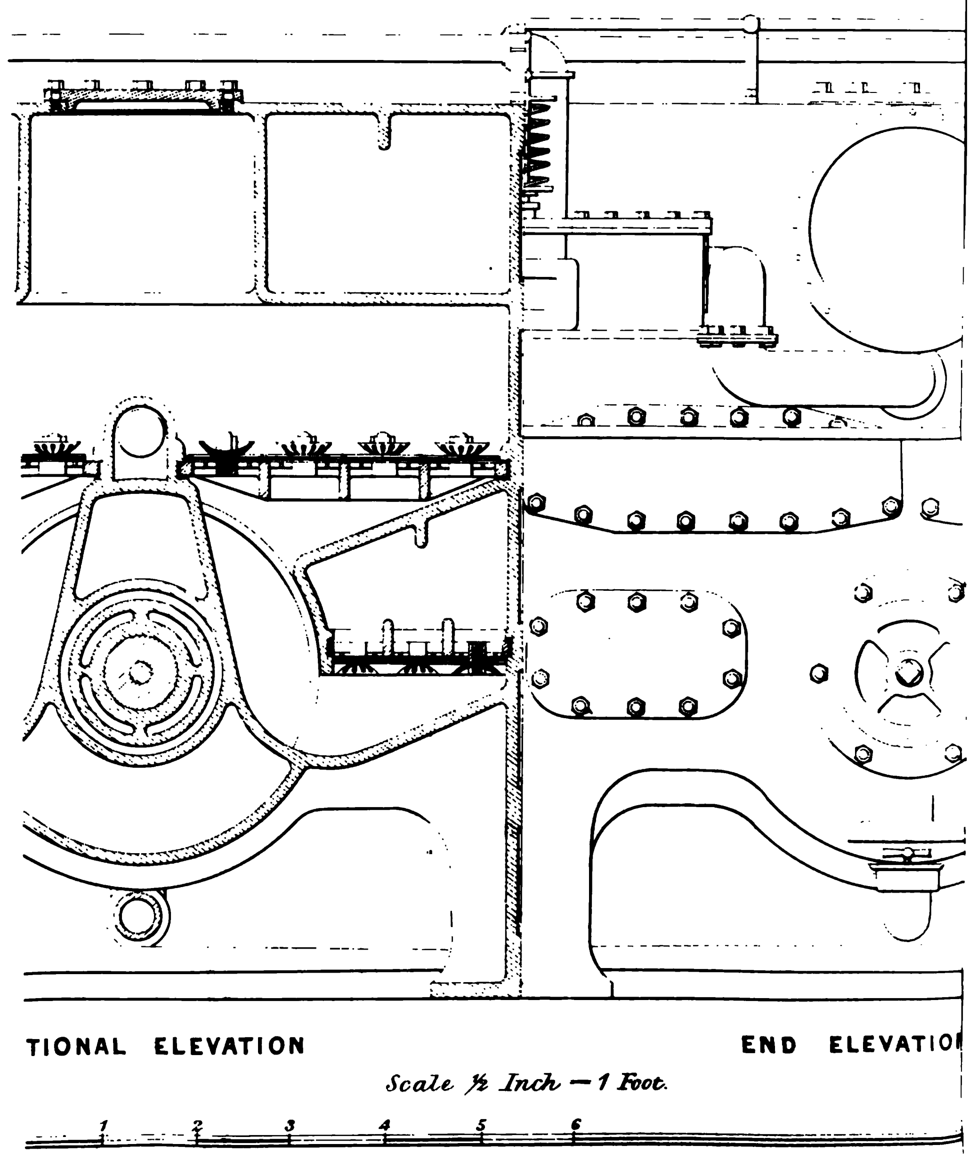
N COMPLETE.

12 Feet

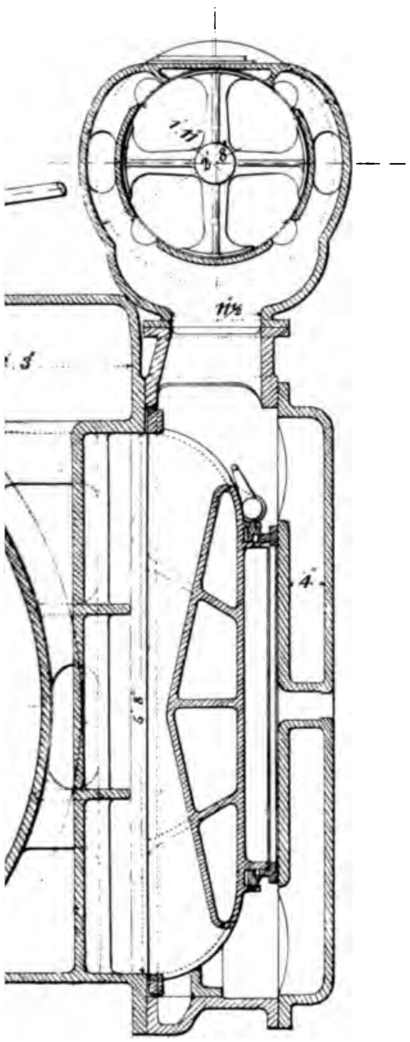
ELEVATIONS OF CONDENSERS, OF THE ENGINE

Fitted in H.M.I.C.S. HECTOR.

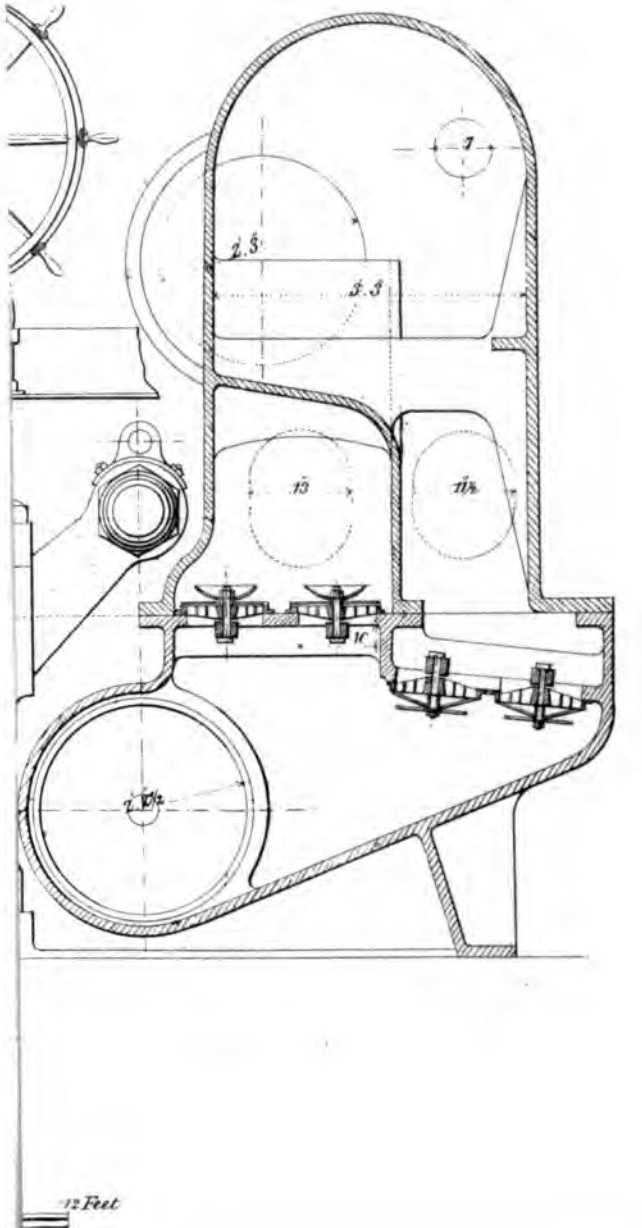
STRUCTED BY MESS^RS R. NAPIER & SONS, ENGINEERS.

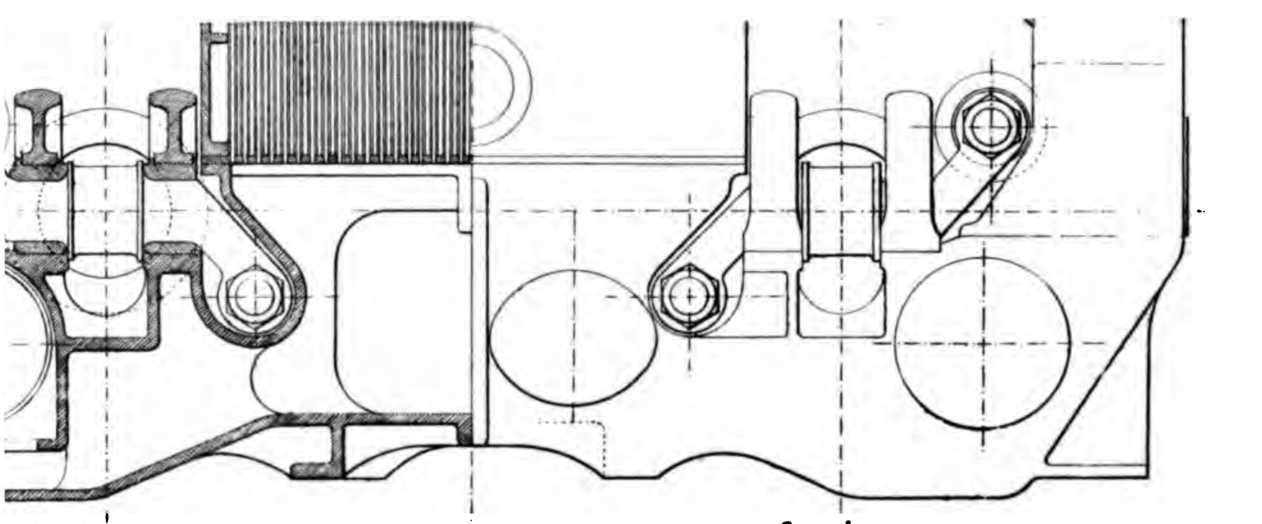


/ELY.



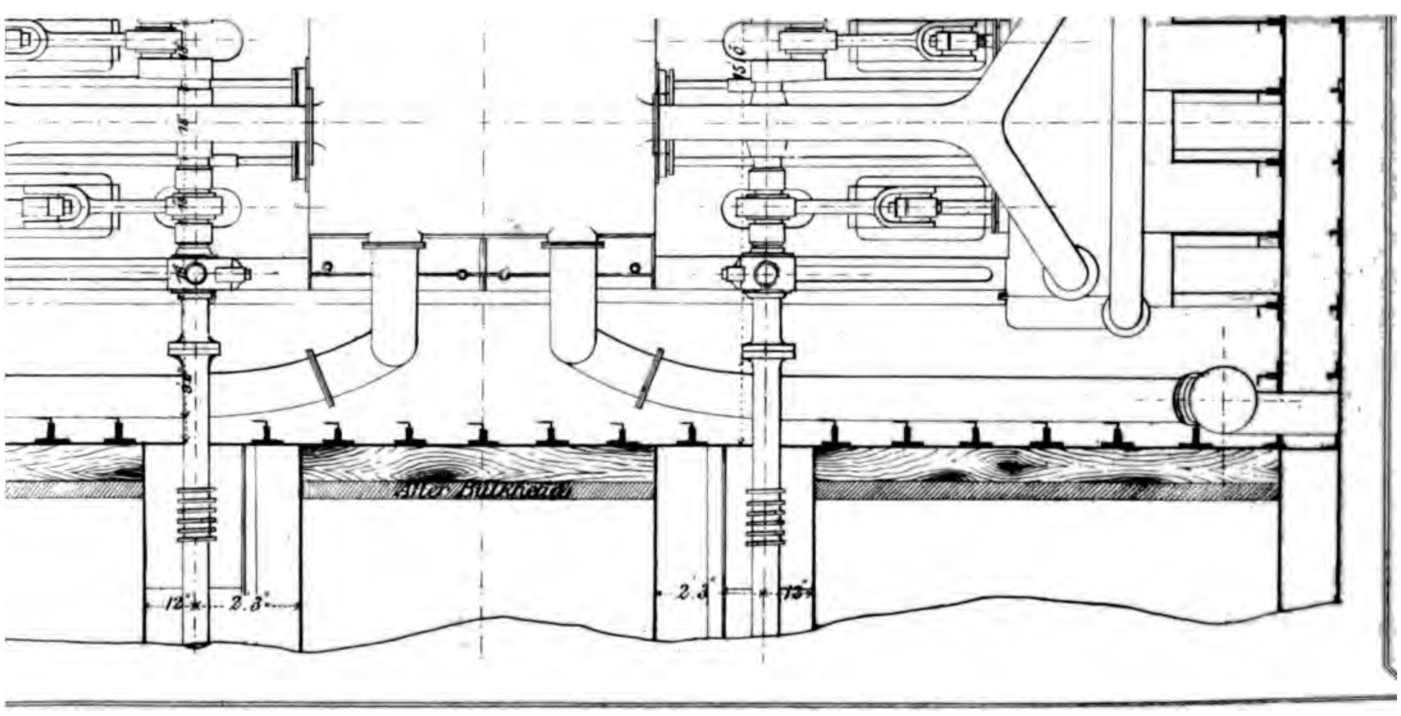
COLLECTIVELY.



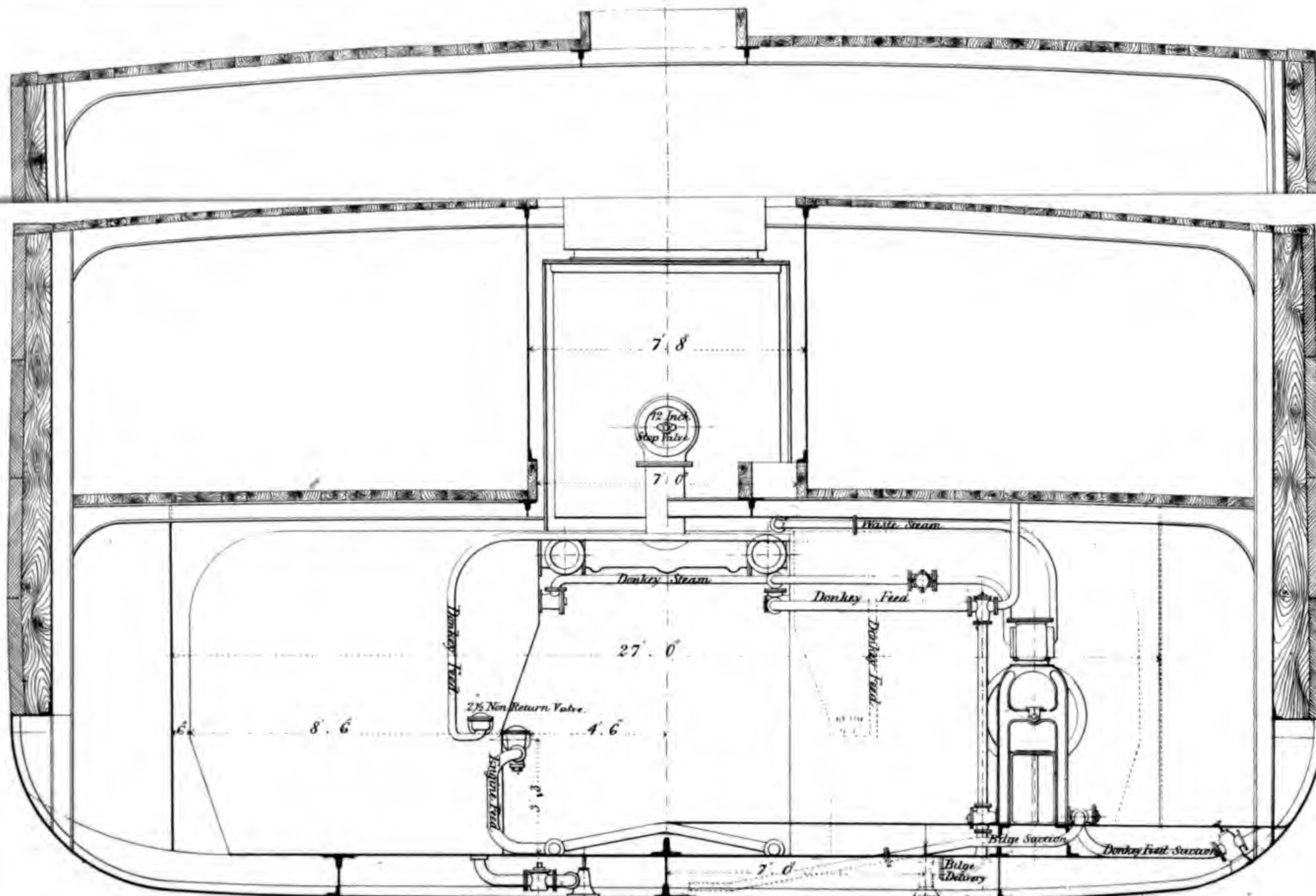


Section

Complete

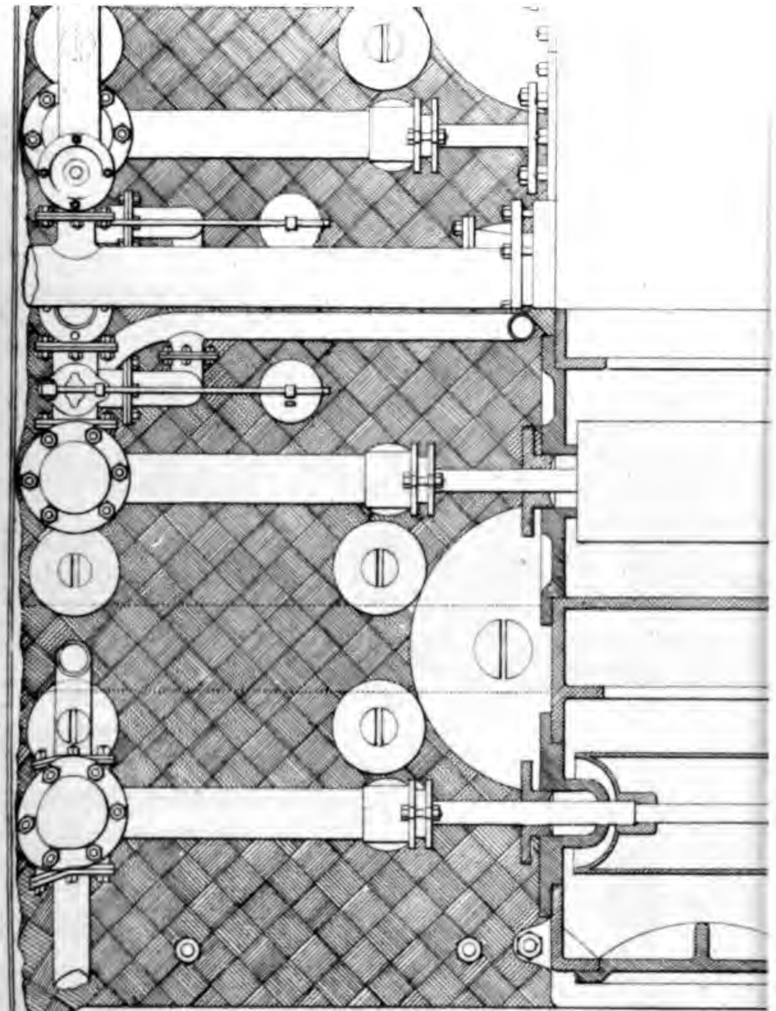


After Bulkhead
12"
2.3"
2.3"
12"



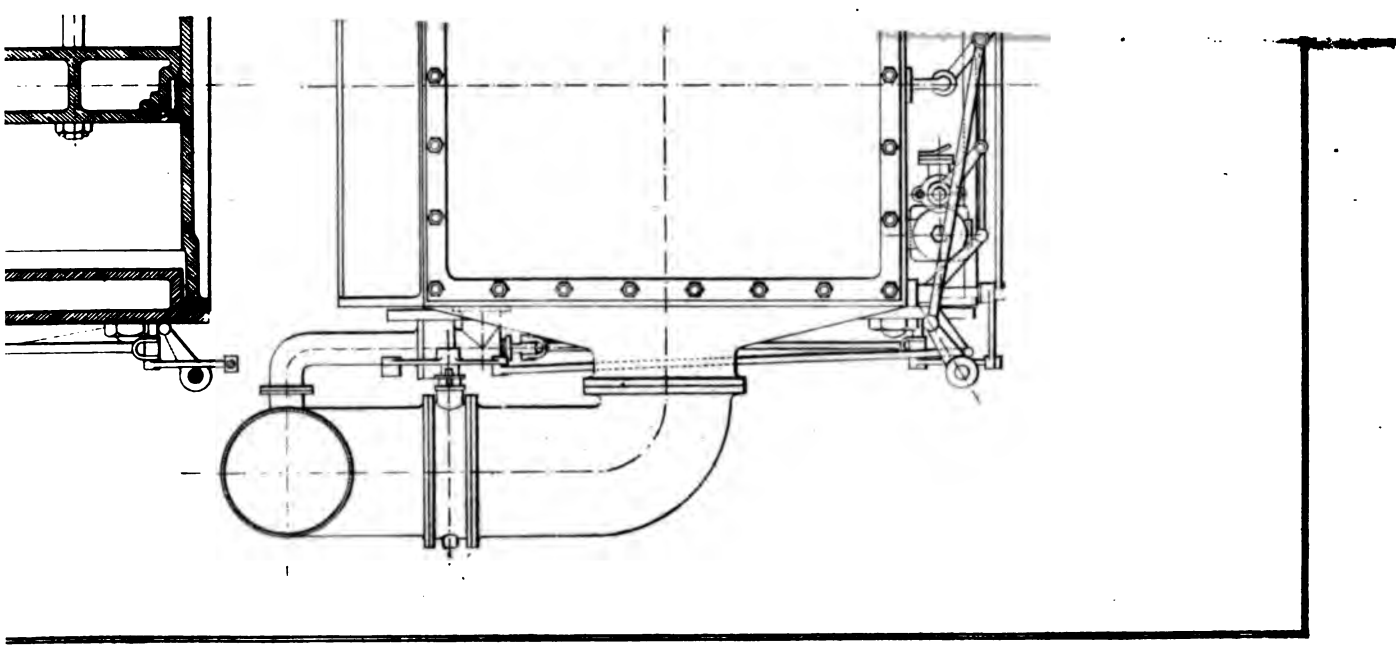
SECTION SHEWING ELEVATION OF BOILERS.

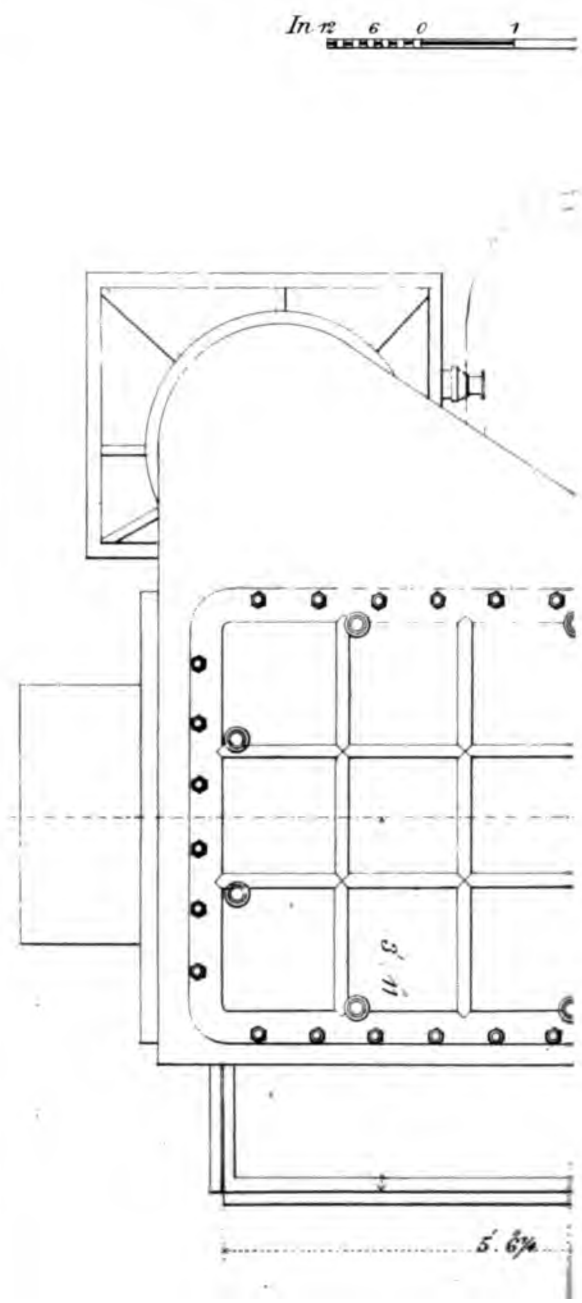
N. P. Burgh, Direx.



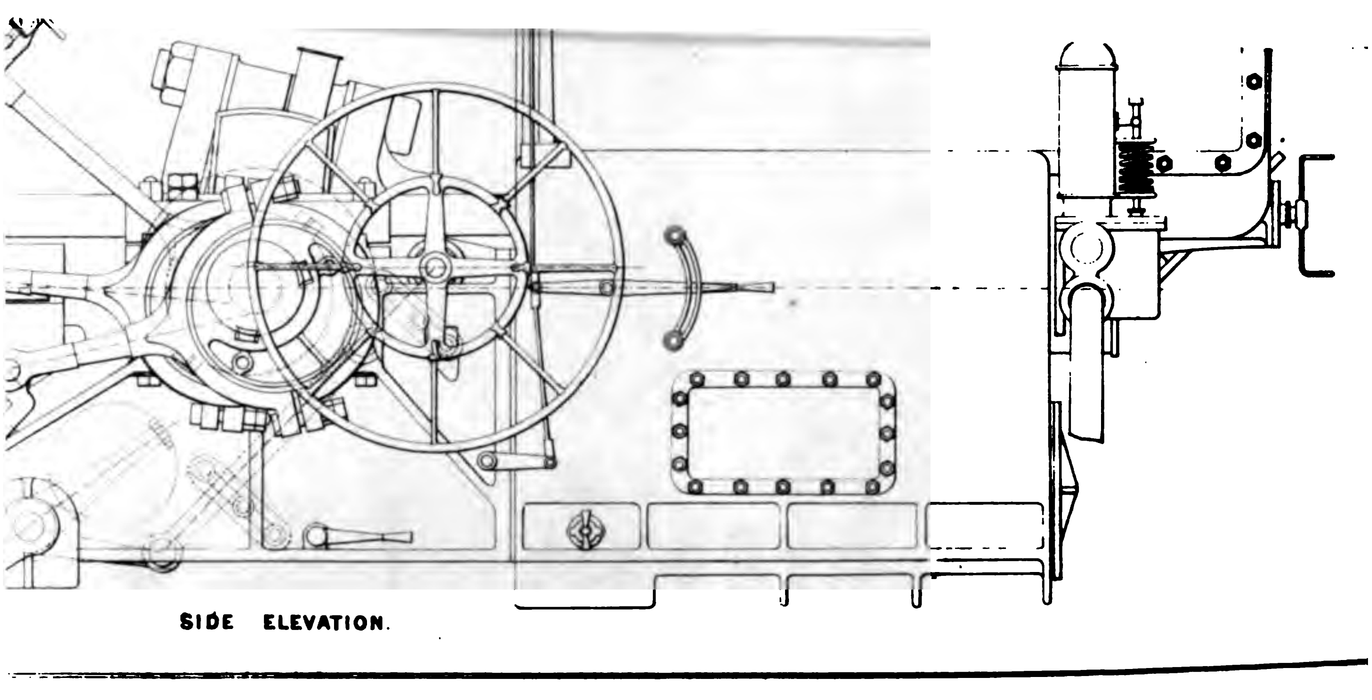
Scale ¾ Inc

Ins. 12 6 0 1 2

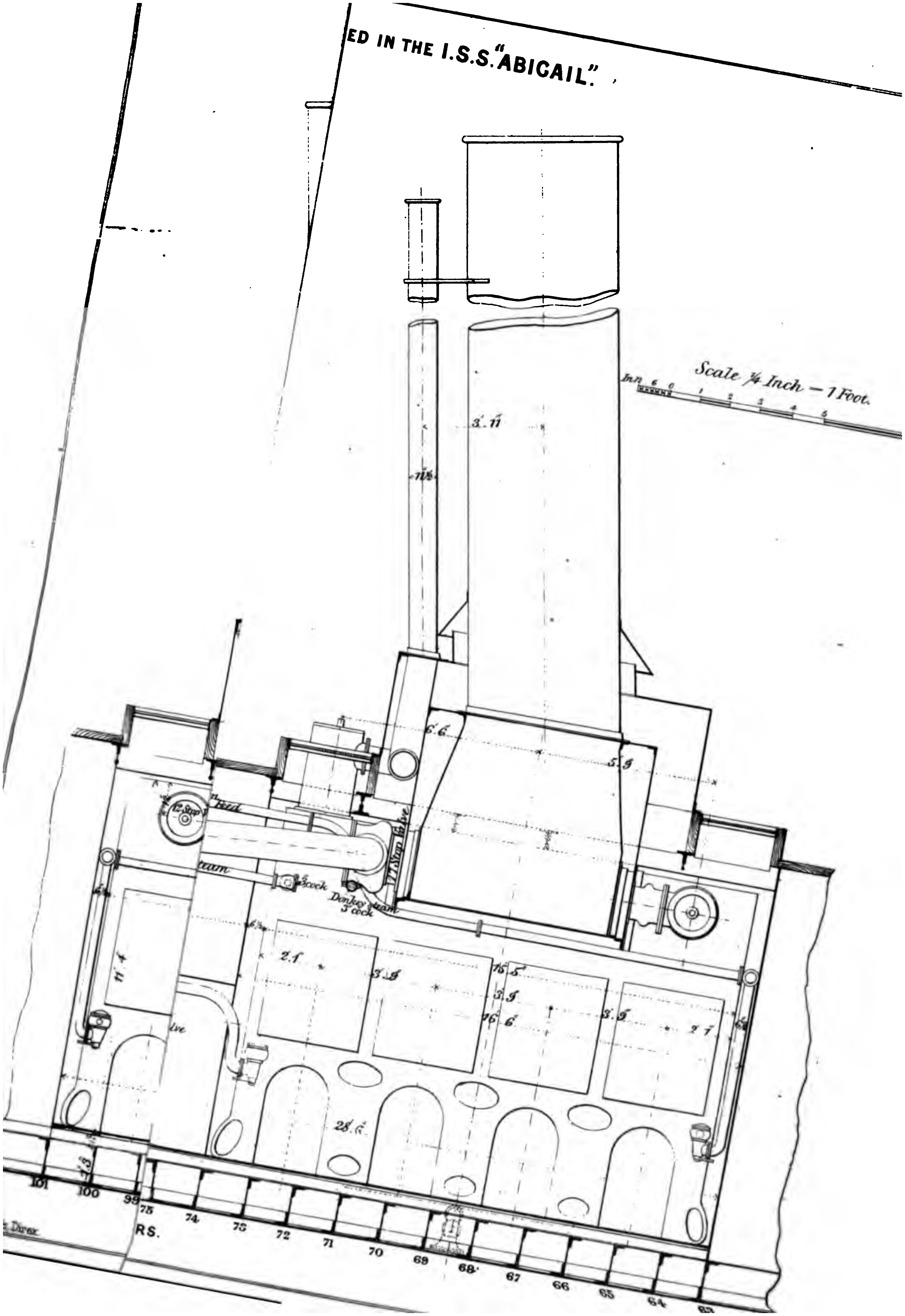


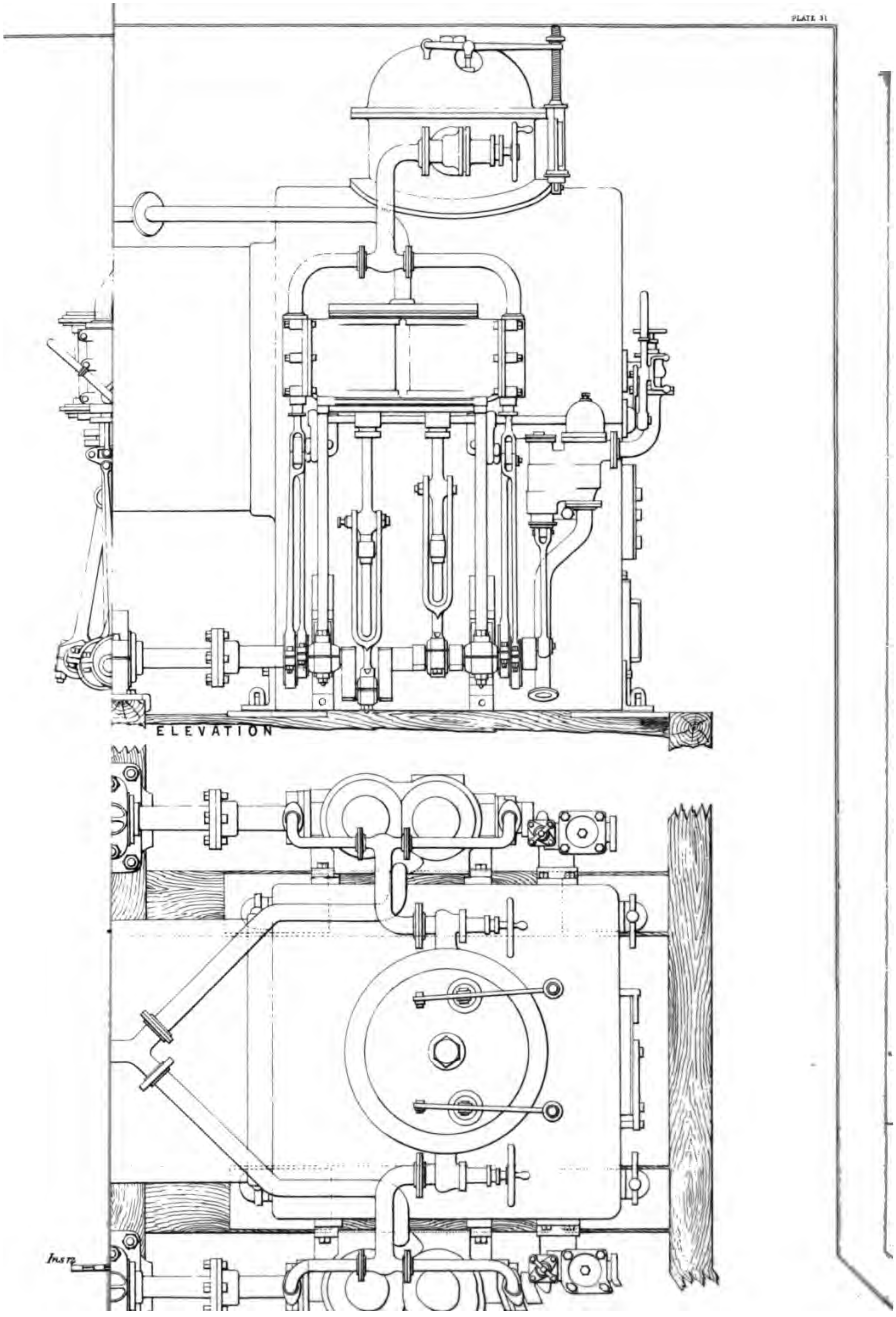


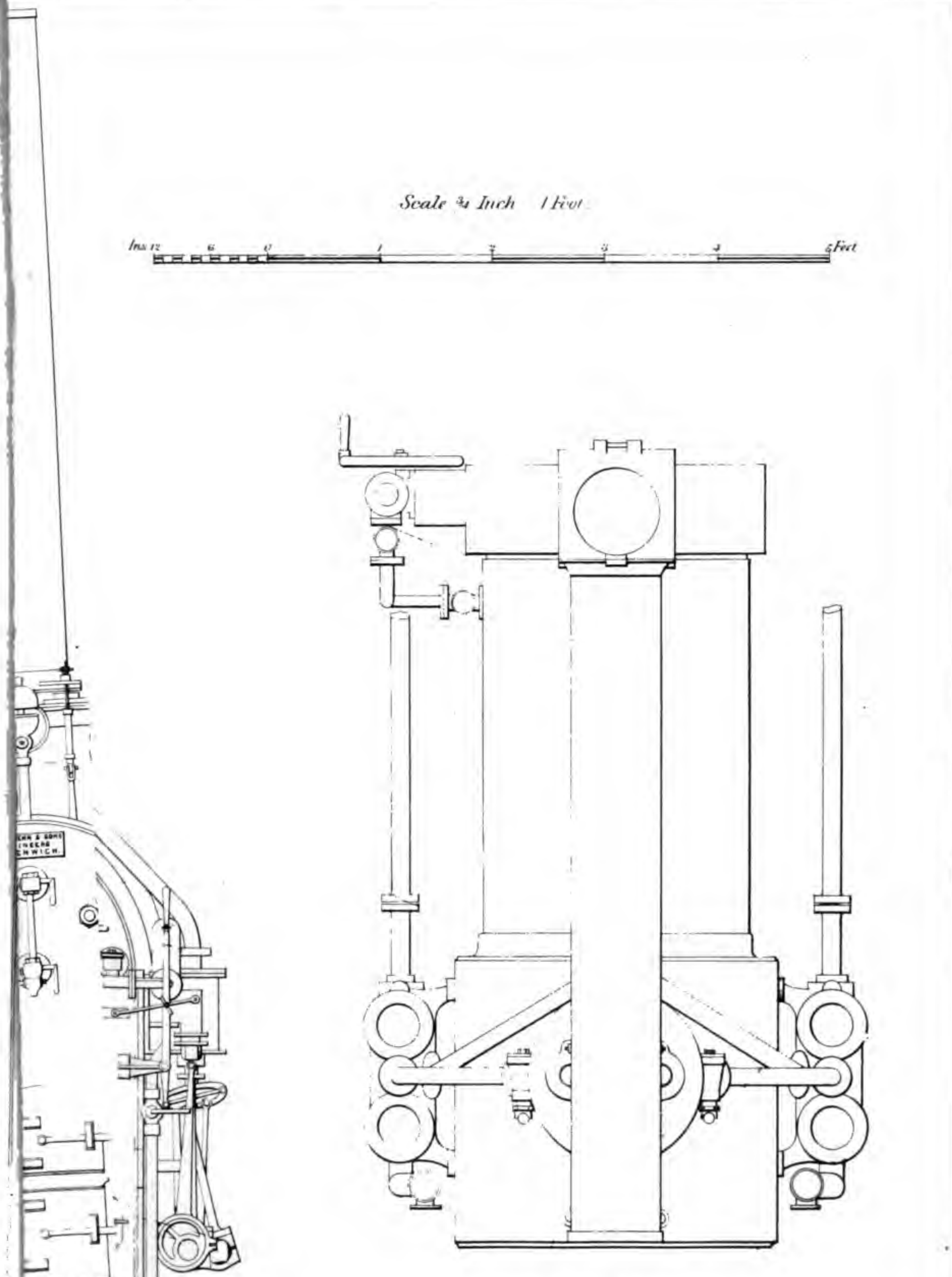
In 12 6 0 1
3' 11
5' 6¾



SIDE ELEVATION.

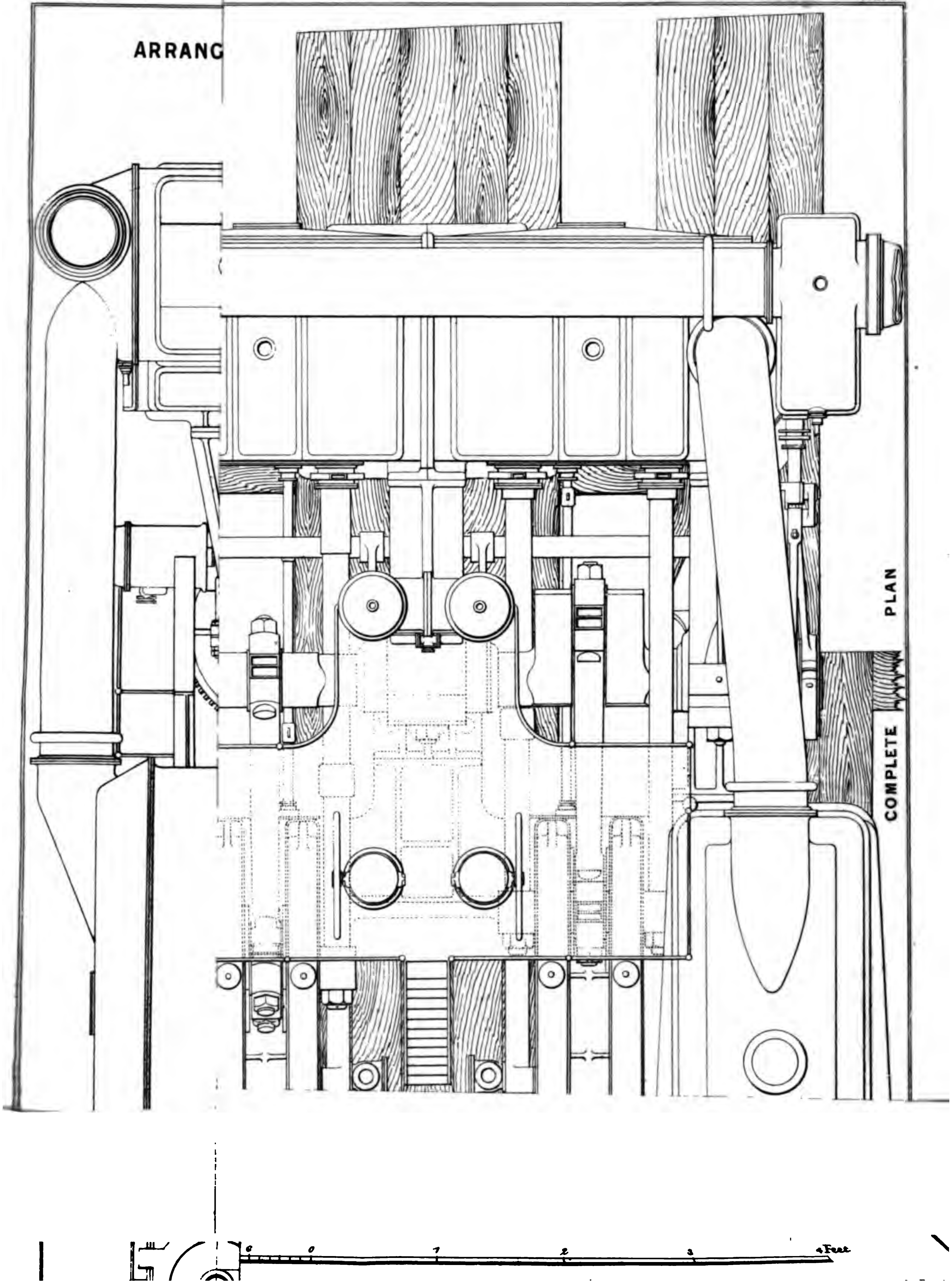

ED IN THE I.S.S. "ABIGAIL".
Scale ¼ Inch — 1 Foot.
12 Stop
Feed
Steam
Stop Valve
Donkey Steam 3" cock
RS.

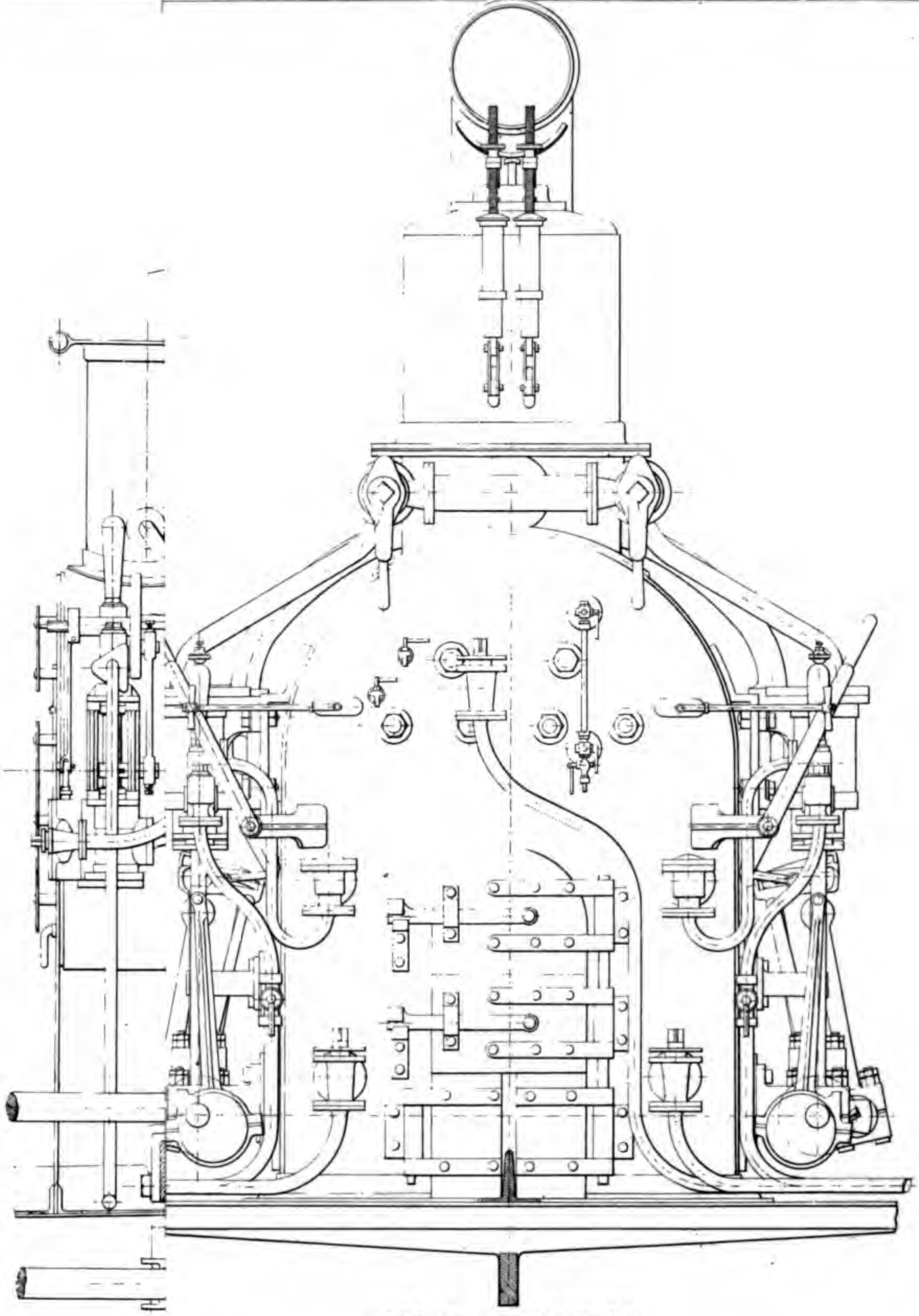

ELEVATION



PLAN

ELEVA.


ARRANG
PLAN
COMPLETE
0 0 1 2 3 4 Feet

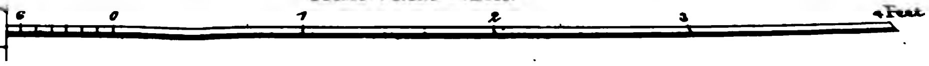


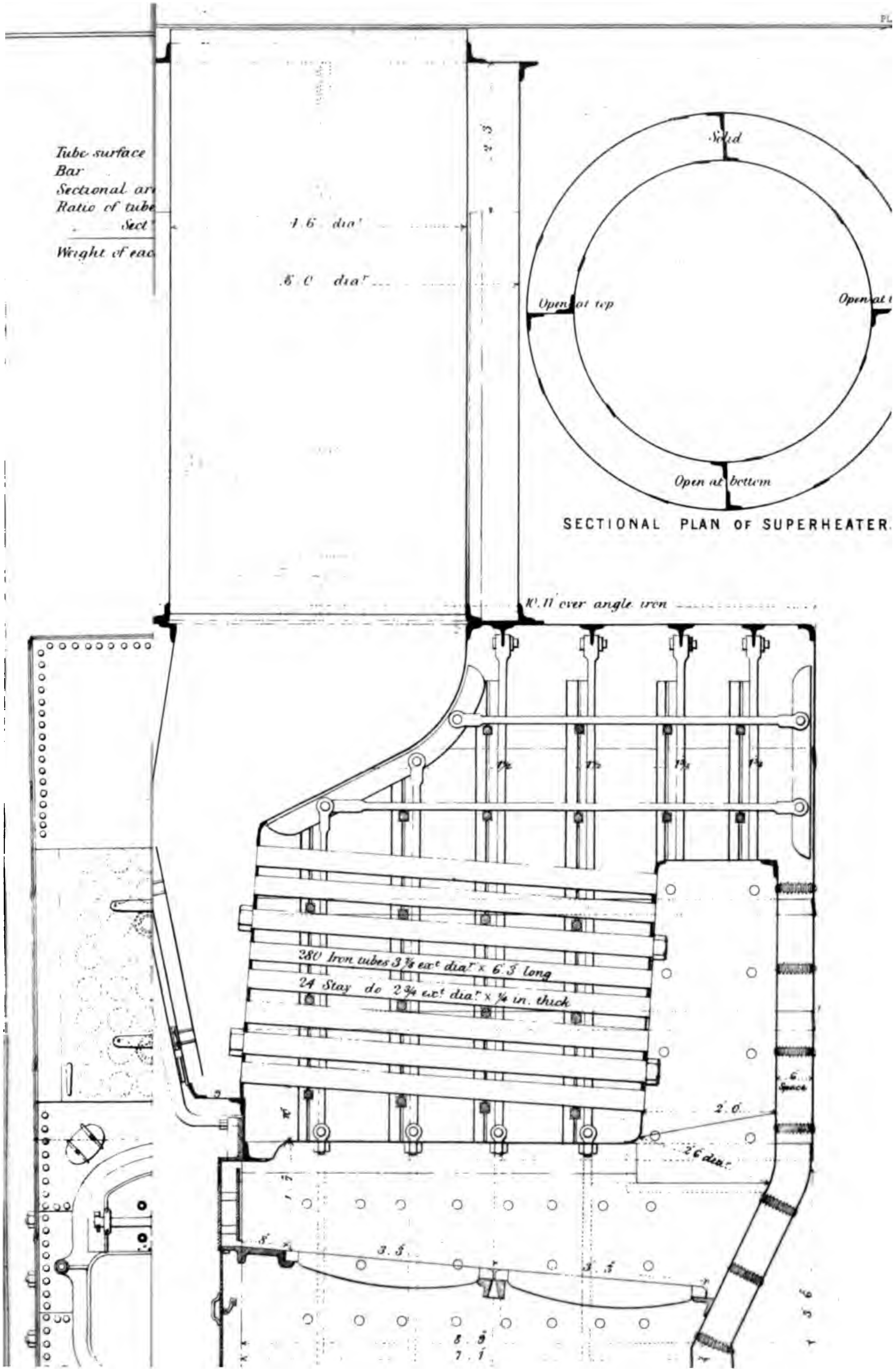
FRONT ELEVATION.

ES AND BOILERS OF TWIN SCREW LAUNCH
FOR "GREAT EASTERN" STEAMSHIP.

G. FORRESTER & Cº ENGINEERS, LIVERPOOL.

Scale 1 Inch – 1 Foot.




Tube surface
Bar
Sectional ar
Ratio of tube
Sect
Weight of eac
4.6 dia
5.0 dia
Solid
Open at top
Open at
Open at bottom
SECTIONAL PLAN OF SUPERHEATER.
10.11 over angle iron
280 Iron tubes 3 1/8 ext dia × 6.3 long
24 Stay do 2 3/4 ext dia × 1/4 in. thick
2.6
2.6 dia
6 Space

APPENDIX.

JUDGING from the comprehensive title of this work, and of this Appendix, an idea may be entertained by many of our readers that it is our intention to go exhaustively into the improvements which have taken place in the world of Marine Engineering since the publication of the last edition of this work, which bears date 1872. And fearing that such an erroneous idea may be entertained, we hasten to explain that the intention of this paper is merely to give an outline of some of the more important changes which have taken place since then; and to cite a few of the most satisfactory examples of increased speed, reduced expense, improved design, or other general advantage; for were we to notice the work of every engineer who has introduced improvements into the marine engine or its connections during these eight years, the space at our disposal would have to be doubled or trebled.

During these years the compound engine has gone on increasing in popularity, and most deservedly so, until it is now the great, standard engine of modern times, and is without a rival to contest its supremacy. In its infancy it encountered difficulties and oppositions of great magnitude; but then engineering was more infantile, and engineers more timid. When, in looking back over bygone years, one thinks of the names of many great engineers, holding the highest positions in the profession, who strenuously opposed its introduction, it is remarkable, as a fact, how signally the principle of compounding weathered the storm, and how great and sterling must have been its advantages to hold its own, and to conquer the prejudices of this same opposition. Of course it is an axiom that that which is correct in science must eventually conquer opposition and take its proper place; but no better argument, if argument were needed to prove the pre-eminent advantages of the compound principle, exists, than that it has overcome great opposition in so short a time, and that it is now unrivalled. And unrivalled it will remain, until a rotary engine, constructed on sound principles and economical, comes into the field of engineering, which most assuredly it will, be it sooner or later. In 1872 the compound engine held the premier position amongst marine machinery, and as it holds the same to-day, we can point to no changes in principle during these years; but in variation of type and dimensions there have been many changes, and in this bustling age, when one engine builder outvies his neighbour and disdains to hold a secondary rank in reputation, we are still likely to have yet many changes of type and many additions to the present dimensions of our engines. It is to the advantage of the world, of science, of capital, that this honest rivalry exists amongst our engine makers and our shipbuilders, and commerce may look on complacently and reap the benefits accruing from their labour. The cylinders of the most popular of all types of engines—that is, the two-cylinder compound inverted direct-acting engine — have gone on increasing in size until a maximum has been reached, and it has been found necessary to introduce a third cylinder when greater power was desired. The considerations which fix the size of a cylinder are practical, for it is found that great difficulties are to be met with in the construction of cylinders of 100 inches in diameter or upwards; they are very heavy and unwieldy to be carried from place to place during the course of their manufacture and machining, the steam portfaces are very liable to crack, and even when the engine is finally erected and on board its vessel, the great weight of all the parts of the engine depending upon, or contiguous to, this cylinder must always be a strong practical objection to the adoption of cylinders of such a size. The practice of building horizontal engines had been considerably more in use up to the year 1872 than since, and almost the last pair of large and important engines built about that time were a mixture of horizontal and vertical. They were fitted on board the S.S. "Montana" for the Guion line, and were built by Messrs. Palmer and Co., of the Jarrow Works, Newcastle-upon-Tyne, this firm at the same time putting a similar pair of engines on board the S.S. "Dakota" for the same line. The high-pressure cylinder of these engines was vertical, inverted, and direct acting, whilst the two low-pressure cylinders were horizontal, working on to their cranks by means of return connecting rods, each piston having four piston rods. The low-pressure cylinders were each 109 inches in diameter, with a stroke of three feet. Steam was admitted to the cylinders by means of a Corliss valve gear somewhat like that used with Mr. John Spencer's engines. It was the "Montana" which had the notorious tubular boiler of Mr. Jordan fitted on board when she first appeared; but as this boiler was found to be an utter failure, it was removed, after a few days' trial, having caused an enormous outlay in the experiment, and boilers of the ordinary circular multitubular type were adopted.

This type of engine, being so extremely complicated in its parts, and so difficult to get at, was never popular,

shipowners and their engineers always preferring a simple arrangement in the engine-room, and this is why the inverted direct-acting engines have gained and maintained a deserved popularity. Of these there are a great many in the field, the general design being alike; that is, they all have vertical columns standing upon a bed or sole plate, the condenser, of the surface-condensing type, being fixed to, or cast in one with, the columns at the back of the engine; between these columns run the crossheads on their guides, and from these depend the connecting rods, whilst from their upper sides rise the piston rods to the cylinders and pistons overhead; the air, circulating, feed, and bilge pumps being driven by means of levers attached to the crossheads by connecting links of a suitable character. Of this type are the great majority of our marine engines, and it is now only on board vessels of the Royal Navy that we may see horizontal or diagonal engines, or those with return connecting rods, their use being necessitated in these vessels by the desire to get the machinery below the water line. In the merchant service it will be observed that the same necessity does not exist, and we are able to get engines of considerable height; a great advantage, in that it allows of our using a long connecting rod, as well as that we are enabled to build our engines so that each part may be easy to get at for examination or for overhauling. The advantage gained from the use of a long connecting rod, is that we greatly reduce the friction on the guides by the fact that the angle at which it works is reduced; for were we able to work with a very long connecting rod, we might reduce the friction to something merely theoretical; but a limit is fixed, and an early one too, to the length of this rod, for it is not convenient nor safe to have very lofty engines, and so the general practice is to fix the length of the connecting rod at from four to five times that of the crank. In types like that of the "Montana," this could not be obtained in the low-pressure engine, and on this account, as well as for the general inaccessibility of the engine and its complication, the type was abandoned.

The inverted direct-acting compound engine will be found on board every ship of that great fleet which has made England's name famous throughout the world—we mean the fleet of what are generally known as "cargo boats." It is also to be found on board some of the largest steamers of the day, notably on board the "City of Berlin," which for a few months more will rank as the largest ship of the English mercantile marine, but for a few months only, until the "City of Rome" takes the leading position. It was not convenient, as we have shown previously, to construct cylinders of so great a size as that of the "City of Berlin's" low-pressure one, and when the same or greater power or speed was required, it became necessary to use more cylinders. Thus we have in the "Orient" three cylinders, in the "Germanic," "Britannic," "City of New York," four cylinders, and in the "City of Rome" we are to have six. In the Appendix to "Compound Engines," and under that title, we have spoken of and described the engines of the "Germanic," "Britannic," "Orient," "City of Rome," etc., and we may now say something of those fitted on board the "City of New York." They were built from the patented design of Mr. Allen, of the North Eastern Engineering Works, Sunderland. The special feature of this engine is, that the bed or sole plate on which it stands is also the condenser, the manifest advantages of having which so low down and under the engine are—1st. The weight of the engine is central, not as in other cases when the condenser is at one side. 2nd. Being so low down, the water from the circulating pumps is enabled to run freely and easily through the tubes; should the circulating pumps give way, the water may be allowed to run through the tubes, and, by the removal of one of the doors, into the bilges, whence the ballast donkey may remove it, if there should be no connection between the donkey and the condenser, which there is in most cases. 3rd. The removal of all lever motion for the working of the pumps, as they are all worked directly from the crosshead, having the same stroke thus as the piston. And 4th. The general simplicity and accessibility of every part of the engine. Each engine is complete in itself; that is to say, should one break down, the other, having air, feed, and circulating pumps attached by rods to its crosshead, will be fully equal to the work of propelling the ship. Again, should one circulating pump break down, the other is of a size which is adequate to the maintenance of a satisfactory supply of water; and on the same principle the air, feed, and bilge pumps are also designed.

The engines of the "City of New York" have, as we stated above, four cylinders, two low pressure of 71 inches diameter, and two high of 40 inches, the stroke being 5 feet. The high-pressure are fixed above the low-pressure cylinders, and a somewhat clever method has been adopted for the examination of the low-pressure cylinder, which we shall briefly describe. In the "Germanic," and her type of engines, the piston rod passes from the crosshead through the low-pressure and up to the piston of the high-pressure cylinder, and as a consequence it would be necessary to remove the upper cylinder from its seating before the lower piston could be taken out were such a thing desirable at any time. Even for an ordinary examination of the low-pressure piston, the space is very small between the two cylinders, as the low-pressure cover must remain on the rod, being only capable of being lifted as high as the bottom of the high-pressure cylinder. In order to overcome this objection the low-pressure piston of the "City of New York" has two piston rods which rise from the crosshead, and are secured to the piston by nuts. Attached to the centre of the same piston is another piston rod, which, passing through a stuffing box in the top cover, goes into the high-pressure cylinder, is fastened to its piston, and passes on through a stuffing box in the top cover of the high-pressure cylinder, so making a capital guide for the piston. Should it be necessary to examine the low-pressure piston, it is only needful to disconnect the high-pressure piston rod from the low-pressure piston, when the cover can be removed altogether, and, if it is desirable, the piston itself may then be readily drawn from its cylinder. The advantage of this arrangement of Mr. Allan's makes itself evident, and in a somewhat different form it has been adopted in the only engines since designed, wherein the high-pressure cylinder is above the low, those of the "City of Rome."

The slide valves of the "City of New York's" engines are driven directly from the crank-shaft, expansion valves being fitted into the high-pressure valve chests; the weight of the slide valves is sustained by having small

steam cylinders fixed above the tail rods of the valve spindles, in which work small pistons fitted to these tail rods. The condenser is furnished with a great number of brass tubes through which the water is circulated, and these tubes are fixed to brass tube plates at either ends by means of wooden ferrules, which are tightly driven in, and which never give any trouble. There is little doubt that this simple plan for securing the tubes is far superior to all the patented and elaborate and costly methods which have from time to time been popular.

The accompanying diagrams have been taken from the after engines; those from the forward ones being almost identical, it is needless to show them. The following are the particulars:—

Fig. 260.

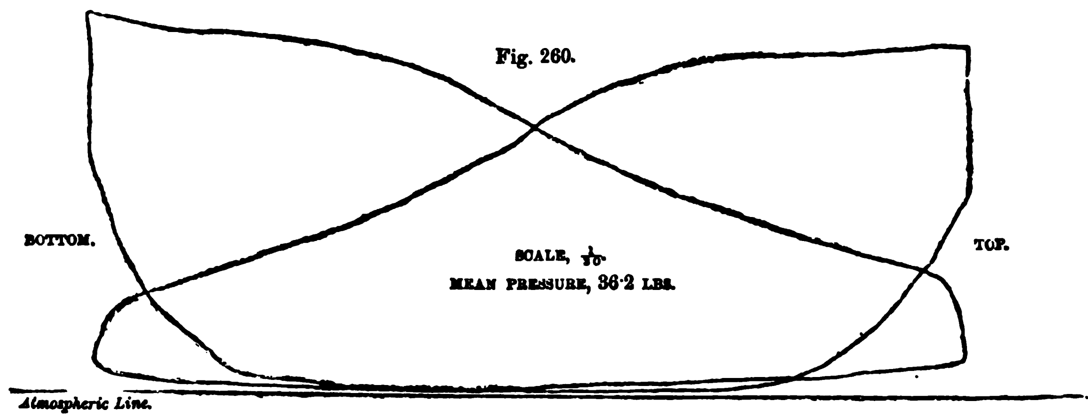
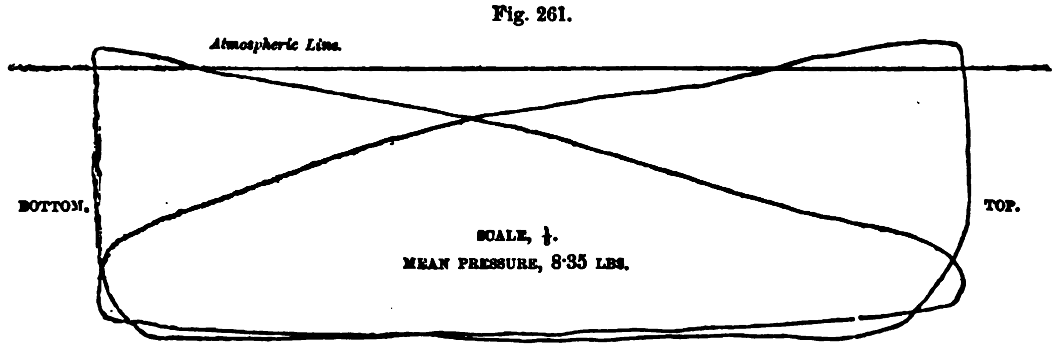


Fig. 261.



DIAGRAMS TAKEN FROM THE AFTER CYLINDERS OF THE S.S. "CITY OF NEW YORK."

Steam Pressure by Gauge	70 lbs.
Stroke	5 feet.
Revolutions	52 per minute.
Vacuum	27½ inches.
Cut off by Expansion Slide	32 "
Indicated Horse Power, Low Pressure	521
" " " High "	717
Total " ...	1238
Total I. H. P. for both Engines ...	2476

During the last ten years great strides have been made in the reduction of the fuel consumption, and now we may set down the general average figures to be 2¼ lbs. of coal per horse power per hour, and from this there will be but little variation. In some cases the expenditure of fuel is still less than this, and no doubt we could mention half a dozen steamers whose engines work as low as at the rate of 1·6 to 1·8 lbs. coal; but, to use inclusive and general figures, we may set down with justice the consumption in the great majority of boats at 2¼ lbs. of coal per horse power.

The most economical engines yet fitted to any ships are those recently sent away from the yard of Messrs. Wigham Richardson and Co., of Newcastle-upon-Tyne, on board the S.S. "Ville d'Oran" and S.S. "Ville de Bône," in each of these steamers Cookson's Hartley coal being used, and when travelling at the rate of 14½ knots, with stop valves full open, the full consumption has been at the rate of 1·517 lbs. If an allowance be made for Newcastle coal, the rate of consumption will be brought below 1½ lbs. per horse per hour, and this is probably a lower consumption than has been attained with any other engines.

Just twelve months ago the French Government solicited tenders for the postal service between Marseilles and Algiers; the lowest of these tenders being sent in by the Transatlantique Company of Paris, they were subsidized for the service. This company, being otherwise engaged largely in inter-oceanic communication, were obliged to order at once a large number of new vessels to be built. Ten of these were decided upon to be fit for either Transatlantic service or Mediterranean, and were ordered accordingly. Last November the orders for

these boats were placed with some of our most eminent ship and engine builders—Messrs. Inglis, Elder, Caird, and Wigham Richardson; the terms of the order being that they were to be finished in England by June 1st, the mail contract beginning a month later. Time was so well kept by the builders, that half the number of boats were finished by the specified time, and the remaining within four weeks of the date fixed upon.

The engines of the "Ville de Bône" and the "Ville d'Oran" are of the double-cylinder compound type, the high-pressure cylinder being 42 inches diameter, the low-pressure 80 inches, and the stroke being 48 inches. There is an expansion valve on the back of the high-pressure slide valve, and both high and low pressure valves are fitted with balance cylinders. The condenser is fitted with brass tubes $\frac{3}{4}$ inch diameter, giving a cooling surface of 4000 square feet; they are tinned as a prevention of corrosion, and are secured to brass plates at both ends by means of screw glands with cotton packing. The water is forced through the tubes by one of Gwynne's centrifugal pumps, and that the feed water may be very hot, the circulating water is forced through the top nest of tubes first, coming back through the lower ones, and so overboard. Steam is supplied by two large double-ended boilers, having twelve furnaces in all. The mail contract speed for these vessels is 12 knots, but the company required a trial speed of 13½ knots, with which, and more, they were accommodated by the separate builders. The engines above described have been designed by Mr. John Tweedy, manager of Messrs. Wigham Richardson's Engine Works, and the ships, which are most sumptuously and elegantly fitted, are from the design of Mr. Christie, one of the partners in the firm. The "Ville d'Oran," which was first at her station, has given the most entire satisfaction, and has attained a speed on several runs of upwards of 16 knots. Amongst the most notable improvements with which both of these vessels are fitted, we may mention centrifugal pumps and ejectors for taking the water from the bilges, Nichol and Donkin's steam steering gear, and Durham's admirable governor.

Of late years the world of engineering has been startled by the gigantic designs which have come to light and ripened; the "Orient" and "Gallia" led off the dance, and then, in quick succession, come the "Arizona," "Servia," "City of Rome," and "Alaska." One of these belongs to the Orient Steam Navigation Company, one to the Inman Steam Ship Company, two to the Cunard Line, and two to the Guion Line. The Cunard Company exhibit great enterprise in their undertakings, and have a fleet at their disposal second to none, representing considerably upwards of 180,000 tons of steam shipping, which is propelled by 50,000 horse power. The Guion Line Company are determined to hold their position as having the fastest ships across the Atlantic. The "Arizona" has made the fastest passage on record, and with the "Alaska" they mean to beat even the "Arizona." The Inman Company intend to maintain their position as having the largest and most comfortable ships, and for speed they mean to make a bid for the leading position, even against the Guion Line.

The following list of some of our greatest steamers will doubtless interest many of our readers, as they will be enabled to make ready comparisons between already well-known ships and those that are to be. There is nothing calling for special remark, in this place, concerning any of these steamers mentioned, if we except the S.S. "Servia," being built for the Cunard service. This steamer must, however, be looked upon largely in the light of an experiment, as she is to be constructed of steel plates. True, steel has been used for the building of ships, but it is still a little distrusted for the purpose. The first steel ocean steamer built was the "Kinfauns Castle," for the Cape mail service. Other merchant steamers of a small size

PARTICULARS OF SHIPS.

Name of Ship.	Line.	Name of Owners.	Length.	Breadth.	Depth.	Gross Tonnage.	Cylinders. H. P.	Cylinders. L. P.	Stroke.	No. of Furnaces.	I. H. P.	Speed.	Steam Press.	No. of Boilers.
			Ft.	Ft. In.	Ft. In.				Ft. In.			Knots.	lbs.	
Adriatic	White Star	Ismay, Imrie & Co.	438	40 8	31 0	3888	Two of 41 in. dia.	Two of 78 in. dia.	5 0	28	3900	14		
City of Richmond	Inman ..	Inman S.S. Co.	441	43 3	34 1	4607	One of 76 in. dia.	One of 120 in. dia.	5 0	30	4000	14	65	10
Bothnia	Cunard ..	Burns & McIver	423	42 0	34 6	4535	One of 60 in. dia.	One of 104 in. dia.	5 6	24	3500	13½		
Arizona	Guion ..	L'pool & Great Wstrn. S.S.Co.	450	45 4	35 10	5146	One of 64 in. dia.	Two of 90 in. dia.	5 6	39	6000	15¼	90	7
Orient	Orient ..	Anderson & Co.	445	46 0	35 0	5388	One of 60 in. dia.	Two of 85 in. dia.	5 0	24	4800	14	90	4
Gallia	Cunard ..	Burns & McIver	430	44 0	36 0	4808	One of 60 in. dia.	Two of 84 in. dia.	5 5	24	4700	14		8
City of Berlin ..	Inman ..	Inman S.S. Co.	489	44 0	34 10	5409	One of 72 in. dia.	One of 120 in. dia.	5 6	36	5000	15	75	12
Germanic ..	White Star	Ismay, Imrie & Co.	455	45 0	33 6	5008	Two of 48 in. dia.	Two of 83 in. dia.	5 0	32	5000	15	75	8
Servia	Cunard ..	Burns & McIver	500	50 0	37 0	7500	One of 72 in. dia.	Two of 100 in. dia.	6 6	38	7500	16		
City of Rome ..	Inman ..	Inman S,S. Co.	546	52 0	38 9	8300	Three of 43 in. dia.	Three of 86 in. dia.	6 0	48	8300	17	90	8
Alaska	Guion ..	L'pool & Great Wstrn. S.S.Co.	500	50 0	39 0 about	7700 about	One of 68 in. dia.	Two of 100 in. dia.	6 0	54	9000 about	17½ about	100	9

have been built of steel, notably an exceptionally fast one running between Newhaven and Dieppe, but the "Kinfauns Castle," which began its work early this year, is the first completed example of the application of steel in the construction of great ocean-going mail steamers, and her performances are being eagerly watched by many engineers. The "Grantully Castle" of the same line is a sister ship of the "Kinfauns Castle," with the exception that she is built of iron, and a fair field for comparison is here established.

We have now gone into the subject of the general type, as well as of the particulars, of some of the most famous ships which have been built during these ten years, and we shall bring our notices of engines and ships to a close with a few of the leading features in connection with Mr. Loftus Perkins' engines, and with those of the "Livadia."

The principles of construction and science upon which Mr. Perkins has worked are so much removed from those in ordinary use, that they have not become popular, and we only find his engines and his now well-known boiler on board very small vessels. No doubt a certain amount of prejudice opposed this engine, and prejudice should be the last feeling in an engineer's mind. As an example of this engine, which may justly be termed a specialité, we shall take for our notice those fitted on board the steam yacht "Anthracite." They are inverted and direct acting. Two cylinders are used, the after one being the high-pressure, the forward the low-pressure. The after or high-pressure cylinder has within it two diameters of bore, the upper or smaller one being the high-pressure proper; it receives its steam during the first half of the down stroke. The lower, the larger diameter, is the intermediate cylinder, and does its work during the up stroke with the steam which made the preceding down stroke in the high-pressure cylinder, and having finished its work in this intermediate cylinder, it is exhausted for use in the low-pressure, which is of the ordinary double-acting kind. Steam is admitted into the high-pressure and intermediate cylinder by means of three lifting, double-beat valves, worked by eccentrics on the main shaft; and into the low-pressure cylinder by means of the ordinary slide valve, on the back of which is an expansion slide, both of which are worked directly from the main shaft. "The surface condenser is composed of a number of close-topped galvanized wrought-iron tubes, standing vertically from a tube plate, and having within them smaller tubes open at both ends, and proceeding upward from a lower tube plate, so that the water from the sea passes up through the central tubes, and down the annular spaces to the inlet of the circulating pump." The air and circulating pumps are worked by beams, one from the low-pressure piston rod, and one from the other; whilst the feed and bilge pumps are worked directly from the crossheads of the air and circulating pumps. The dimensions of the cylinders are—high-pressure, $7\frac{3}{4}$ inches; low-pressure, $22\frac{13}{16}$ inches; and intermediate, $15\frac{13}{16}$ inches; the stroke of both cylinders being 1 foot 3 inches. Mr. Bramwell, to whose report we are indebted for these particulars, made a very careful trial of these engines last June, the indicated horse power being about 85, the steam pressure 370 lbs. by gauge, and the coal consumption 1·83lbs. of coal per horse-power per hour.

The boiler is the well-known Perkins' boiler, which is constructed entirely of tubes, connected by vertical thimbles at regular distances, and is capable of supplying steam at a pressure of 500 lbs. per square inch. It is supplied with distilled fresh water, there being a still, with its steam pipe attached to the condenser, for the purpose of keeping up the supply, constantly at work. The fire grate area is about 15 square feet, and the revolutions of the engines $130\frac{1}{2}$ per minute. Mr. Bramwell tells us that "they worked with the most remarkable smoothness and regularity" during the twelve hours' trial which he conducted, "that they were thoroughly steady, no straining or racking of any kind; in fact, it is impossible for any pair of engines to have worked in a more thoroughly satisfactory manner, so far as all external working was concerned, and there were no sounds to lead to any other belief than that of all being satisfactory within." The "Anthracite" has since made a voyage to America and back. While in the States, her machinery was tested by a Board of Naval Engineers for twenty-four consecutive hours. The consumption of American navy coal was at the rate of 2·71 lbs. per horse per hour.

Before closing our notice of these engines, we may mention the fact of there being an enormous waste of steam pressure between the boiler and the high-pressure cylinder. According to the diagrams taken from the engine by Mr. F. J. Bramwell, we notice that though the pressure by the steam gauge is 370 lbs., the initial pressure of the high-pressure cylinder is below 200 lbs., so that during the trials of the "Anthracite" an enormous amount of throttling took place, and whether this is to be always the case is unexplained.

We come now to what is perhaps the most wonderful piece of naval architecture and engineering which the world has seen for many years, and in the construction and design of the Emperor of Russia's new ocean-going yacht, the "Livadia," a great field has been opened up for surmise, and for speculation as to her success. It was, of course, necessary to construct a vessel which should cause but little uneasiness to His Imperial Majesty, in the way of rolling or tossing, and of a design which would allow of accommodation suitable for an emperor being provided; and, so far as we may yet be able to judge, these conditions have been fulfilled. The main body of the ship is in shape exactly like a huge sole or turbot, and is 235 feet in length, with a breadth of 153 feet, and a draught when ready for sea of only 6 feet 6 inches. In this part of the ship are the engines, coals, boilers, and general stores. It is constructed of steel in order to insure lightness, and is perfectly flat as to its bottom, which is double, and which is 3 feet 6 inches in height at the centre. This double bottom is divided into 40 water-tight compartments. At the sides two vertical bulkheads run right round the ship, the distance between which and the outside skin is subdivided into 40 other compartments. The whole of this lower portion of the structure has been built of great strength, so as to resist the strains which will be thrown on it by the motion of the waves, as well as by the working of the engines. Upon this main body of turbot shape a large superstructure has been constructed, which bears the form of an ordinary ship, and is, like the main body, built of steel. In this are situated berths for the crew forward, and for the officers aft, whilst the palace

above contains the imperial apartments and cabins for the suite. The palace is not so wide as the steel superstructure, so that all around it a gallery is formed, and as the roof is carried out to the same width as the lower structure, an awning is provided, which shades from sun and rain the lower story of the palace, and at the same time adds to the width of the promenade overhead. At the forward end of the lower story are situated the apartments for the Emperor, and at the after end the suite is located. On the awning deck, and just over the Emperor's rooms, is a grand reception saloon, both large and lofty, which is furnished in the most sumptuous fashion; and on the same deck, but right aft, accommodation has been provided in another deck house for the Grand Duke Constantine, and for the captain commanding the yacht. Looking at this curious vessel, one is struck by the great height of the decks, which seem to tower above the hull. At the top of all is the grand reception saloon and the Grand Duke's apartments, which are situated some forty feet above the sea level. And truly does one recognize the necessity of having an enormous base, in order to insure stability. With such a huge area as this for the wind to act upon, and with so light a draught of water, it is open to very grave conjecture, whether the "Livadia" will ever be able to steam well against a head wind; a few months more will settle the matter, but in the mean time we cannot help surmising that she will be a vessel of a peculiarly "fine weather character," and that neither big seas nor strong winds will at all suit her. She is designed for and intended to be run in the Black Sea, and so will not be obliged to face waves of any great size.

The propelling power is provided for by three engines, each of which drives a propeller of its own. The engines are of the three-cylinder type, the high-pressure cylinder being 60 inches diameter, each of the low-pressure 78 inches, the stroke being 3 feet 3 inches; and the entire force which will be used in driving this vessel through the water at a speed of 14 knots is 10,500 horse power. The propellers are made of manganese bronze, a whitish-grey metal, which is found to be both light and strong; and in the construction of the engines and boilers steel has been very largely used, with a view to reduce the weight, whilst maintaining the strength at a high degree of efficiency. We have said that the "Livadia" is to travel at the rate of 14 knots per hour, and this velocity she exceeded considerably on her trial trip. As at this speed she will consume 200 to 250 tons of coal per day, whilst only possessing a displacement of 4000 tons, she can hardly be looked upon as a wonder of economy, for we have very many steamers, of a considerably greater displacement, making their voyages at a rate of 12 to 16 knots with less than half this consumption. In fine, the "Livadia's" only claims to attention seem to be centred in the facts of her steadiness at sea, and of the light and loftiness of her state apartments, and there is very little in her construction to warrant the sanguine belief entertained by some of the engineers and naval architects connected with her, that she is the pioneer of the coming style and type of vessel for inter-oceanic communication.

We have purposely omitted to say anything concerning the boilers for this yacht, for the simple reason that it has been so difficult to obtain any reliable facts. This much, however, we have been able to find out. She was to have had ten main boilers, and they were to have been of steel, and to be able to withstand the test pressure of 150 lbs. Six of these were finished, being made of plates of Cammel's steel, before one of them was subjected to the hydraulic test. On the pump being applied to the first, and when the pressure had reached 120 lbs., the solid steel plate gave way, the split running some three feet along the plate and then turning out at one of the rivets. The test was applied to the other boilers, and three more of them gave way before the test pressure had been reached.

The boilers were 14 feet 3 inches diameter by 16 feet long. The plates, being as above stated made by Messrs. Cammel, on the Siemens-Martin process, were $\frac{3}{4}$ inch thick, and the steam pressure used was to have been 75 lbs. Of course, after the failure of the shell plates, they were all condemned, and were replaced by shells manufactured by the Steel Company of Scotland; and with these new shells some of the boilers which have been finished, were tested most satisfactorily. Such are the facts, and the failure of the first plates will be a lesson to engineers generally, and to shipowners, and will go far to strengthen the arguments of those who strenuously oppose the use of steel for the purposes of boiler construction. But we are not justified in condemning altogether steel and all steel for this reason, for there are many different qualities of the metal in the market, of which some at least are undeniably good. At the recent meeting of the Institution of Mechanical Engineers held at Barrow, one of the members, a well-known and experienced engineer, stated that the steel used in the construction of the "Livadia's" boilers "was little better than cast iron." Amongst engineering subjects, there is hardly one which has given rise to more diversity of opinion than the advantages or disadvantages of steel; for whilst some maintain that it is useless and most dangerous to use for boiler construction, others maintain that in its strength, lightness, and safety, it is beyond comparison. On the whole, there has been a very strong feeling against its use, for it is undeniable that steel is much more treacherous than wrought iron. But as experience is obtained better results will be got. Messrs. Hicks, Hargreaves, and Co., of Bolton, have made some 12,000 tons of steel boilers, and express themselves quite satisfied with the result of their work, and this is a large experience. We have seen many boilers constructed of plates made by the Steel Company of Scotland, and all of them have given perfect satisfaction. The plates made by the Zandore Steel Company are also of a high order, and Messrs. John Fowler and Co., of Leeds, a firm very largely experienced in its use, maintain that there "is nothing like steel." Now, as the advantages of using this metal are evident, we must hope that the processes now in use for its manufacture will be improved upon, and that we may be able to obtain steel for boiler construction which can be infallibly relied upon, or as nearly so as good sound testing, which has not yet been adopted, can make it.

In no branch of engineering has so little been done in the matter of improvement, as in that of the design of the marine boiler, the existing type of which is very much

behind its partner, the compound engine. Certain new forms of boilers have from time to time been patented, built, tried, and failed, and though they may themselves have been admirable in theory, and only defective in design or in detail, their failure has frightened shipowners, who prefer to stick to the boiler which has done well for so many years, than to embark in any experiment. Jordan's patent tubular boiler, which was put on board the "Montana" at the commencement of her career, was a daring but unmechanical departure from the beaten path, and turned out, as had been foretold, an utter failure, the experiment costing tens of thousands of pounds. Many designs of boilers have been introduced into the field of engineering since Jordan's, but none have been able to oust the present existing type of circular multitubular boiler from its position. One of the latest and best known of these invaders is that patented by Mr. Loftus Perkins, and which has met with but sorry success up to the present time. It is built up of tubes containing water, which is generated very rapidly into steam of a high pressure, and as the tubes are small in diameter, being only three inches, they may be made so thin that the fire has great effect upon the water contained; the complication of the tubes and the difficulty of keeping them clean are, however, a great drawback to their use. The boiler of this type now being tried on board the steam yacht "Anthracite," is fourteen "sections," or tubes, in width; there are seven rings of tubes in the fire-box, and ten layers of tubes above. This boiler has 633 square feet of heating surface, and 15 of grate. Time and space will not allow of our speaking at more length concerning this or any other of the patent boilers now in the field. As we have said above, we still use the tubular boiler which was first introduced with the adoption of surface condensation in 1844, and which has been increased in strength, as the tendency of successive years has been to increase the pressure. If we are to be obliged to use so heavy and cumbrous a generator of steam, undoubtedly the circular multitubular boiler is the best to suit our most modern requirements, and during the last few years great strides have been made to perfect the workmanship in their construction. Rivet holes are now invariably drilled, when but a few years ago they were punched, and the advantages of this improved mode are, that as the holes are drilled through both plates at once, they must of necessity be fair and true; another great advantage is that the violent shock caused by the punching action is obviated, and the chances of cracking the plate reduced to a minimum. Plates are tightly bolted together before being riveted, and so a good and safe job can be made of this operation, instead of the very defective one usual when the rivets were used to draw the plates together. The edges of plates in some of our best shops are planed before being put together, and so a perfectly caulked joint can be made. We illustrate as an example of a first-class modern marine boiler, one of those built for the S.S. "City of New York." The following are a few of the figures in connection with this boiler:—

	Sq. ft.
Heating Surface of Tubes	1858
" " other Parts	632
Total	2490

	Cubic ft.
Steam Space in Boiler	493
" " Dome	99
Total	593

Shell Plate, 1 inch thick. Rivets, 1¼ inch.
Horizontal Seams, treble riveted; Vertical Seams, double.

At one time the corrosion of marine boilers gave grave cause of anxiety to engineers; but it has been found that there is little need for trouble if boilers are treated well, and carefully washed as often within reason as is convenient, and treated by some of the many methods of opposing corrosion. The use of zinc plates has been very largely adopted, and with complete success, both in the boilers of the merchant and royal navies. The brushing of lime on the interior parts has also been recommended and adopted, as well as many other admirable methods of opposing this pernicious corrosion, which most undoubtedly will go on to an alarming extent if boilers are not carefully attended to and looked after. In the case of a marine boiler there is one absolutely necessary precaution which every engineer in charge of boilers should take; *i.e.* he must, at starting, endeavour to get a thin scale to form over the surfaces of the tubes, by a judicious admission of feed from the sea, and a periodical blowing off from the bottom of the boiler. If this scale is formed no corrosion can take place; if not, the consequences may be very serious.

Before closing this Appendix, we must say a word as to the improvements which have been made in marine governors during these ten years; indeed, it is only within the last three years that any genuine improvements have been effected. Up to that time shipowners were very callous as to the application of a governor, and considered that the engineers of the ship should be quite able to ease the engines, if indeed at all necessary, when racing. But after a deal of persuasion they were induced to believe, not only that broken shafts might be obviated, but also that a considerable economy of fuel might be effected, were a governor employed to check the racing of engines in a seaway. A machine, patented by Mr. T. Silver, and which had been in use for some years on board vessels of the Royal Navy, was adopted for the ships of the mercantile marine, but with very indifferent success; for, having only the momentum of a fly wheel with which to close the throttle valve, it was not at all to be relied upon for the accomplishment of its duty.

Recognizing the defects of existing momentum governors, several inventors applied their brains to the introduction of a more efficient one, with the satisfactory result that three came into the field, being respectively the Dunlop, the Westinghouse, and Durham's velometer, each of which is coming rapidly into general use. The Westinghouse has been very much used, and is very popular in the United States; and as Durham's is rapidly being taken up in this country, we will describe its action by reference to the accompanying sketch.

A is the driving pulley, driving the bevil wheel *B*, which gears into the bevil wheels *C* and *D*, which wheels run on trunnions or arms on the square block *E*, the spindle *F* running loose in the block. The two wheels *C* and *D* gear into a fourth bevil wheel *G*, which is fast on the hollow spindle *H*, which spindle has also the wheel *I* fast on it,

and drives the wheel *J*, which revolves the fans *K*; these fans run through fixed diaphragms in the cylinder filled with water, thus securing extraordinary quickness and accuracy of action. The block *E* has fixed to it, by the pin *M*, the spindle *N*, which passes through the hollow spindle *H*, and has on it a sheave *O*, to which is fastened a chain *P*, which runs through a spiral spring *Q*. *R* is a small steam cylinder, the slide of which is connected with the spindle *N* by the lever *S*; the piston rod *T* of this cylinder is connected by a lever with the throttle valve *U*.

The action of the governor is as follows :—The motion is taken from the engine shaft to the pulley *A*, which through the wheels *C* and *D*, drives the fans in the cylinder; the wheels *C* and *D* have a motion independent of that given by the wheel *B*, and are held in position by the spring *Q*, so that there is always an exact balance between the resistance offered to the fans by the water, according to the speed at which they revolve and the tension of the spring; any variation in the speed of the fans causes the wheels *C* and *D* to alter their position, and in so doing to move the slide of the steam cylinder *R*, allowing a little exhaust on one side of the piston, the movement of which *instantly* alters the position of the throttle valve *U*, and so regulates the speed of the engine with the greatest accuracy. The wheel *W* is for setting the governor to cut off at any required speed, and when once adjusted one revolution per minute beyond such speed is an absolute impossibility.

X is a plan of the bevil gearing.

The whole is firmly bolted on the bed-plate *V*.

We can speak from personal experience of this admirable little machine as to work, for we have had occasion to bless it during many a stormy night in the Atlantic. Some 500 of these governors have now been fitted to steam ships, and all seem to give the most entire satisfaction.

One more improvement in modern marine engineering most certainly deserves mention, and that is the adoption of spring safety valves to our boilers, of which many have been introduced.

The most popular of these, and the one which is now almost universally used, is that of Mr. Adams. It is small and compact, and possesses many advantages over other safety valves, the most important of which are—that it relieves the boilers at once of pressure, and returns to its seat with the least practical loss of pressure; that its relieving power is twice that of the generation of steam; and that it is entirely beyond the reach of being tampered with.

This paper has grown to a much greater length than it was our intention that it should when beginning; but if it has served to indicate to its readers some few of the more important strides which have been made in the marine branch of engineering during these ten years, our wish is answered and the object with which we wrote it is obtained, and we need not therefore apologize for its length.

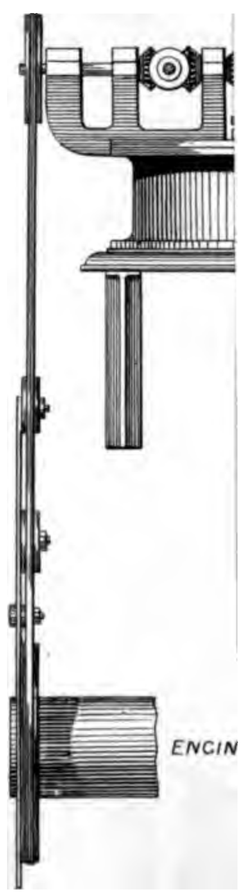
DURHA

HYDRO STEAM

AND

MARINE ENGINE

FIG. I.



W YORK",